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EXPERIMENTAL DEMONSTRATION OF ENERGY STORAGE SUBSTATIONS FOR AIRCRAFT ACTUATION FUNCTIONS

C. W. Helsley, Jr., B. J. Call

LOS ANGELES DIVISION
NORTH AMERICAN ROCKWELL CORPORATION

FINAL TECHNICAL REPORT AFAPL-TR-67-143

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FOREWORD

This is the final technical report covering a Phase II effort on research and development on the flywheel as an energy storage device for use in the performance of aircraft actuation functions. The Phase I effort investigated energy storage concepts for aircraft actuation functions, and exhaustively analyzed all pertinent design features of the flywheel.

This Phase II effort is an experimental demonstration program for the operation of aircraft flight control and utility function simulators using energy storage substations which have flywheels as the principal energy storage element. Both the Phase I and Phase II programs were conducted by North American Rockwell Corporation, Los Angeles Division, under USAF Contract AF33(615)-2971, Project No. 8128, Task No. 2, and USAF Contract AF33(615)-5173, Project No. 3145 and Task No. 314528, respectively.

Initial Phase I effort began 1 July 1965. The period of effort which this report documents is from 1 July 1966 through 31 December 1967. This program was monitored by Mr. R. Smith (APIE-3), of the Air Force Aero Propulsion Laboratory, RTD, Wright-Patterson Air Force Base, Ohio. All effort was conducted at North American Rockwell Corporation, Los Angeles Division, under the direction of Mr. C. W. Helsley, Jr. as Program Manager, assisted by Mr. B. J. Call.

This report is identified as North American Rockwell Corporation Report No. NA-67-964, pending USAF approval. Publication of this technical report does not constitute Air Force approval of the report's findings or conclusions. It is published only for the exchange and stimulation of ideas.

Paul E. Peko, Maj USAF
Br Chief, Propulsion and Power
Aerospace Power Division

ABSTRACT

There are two principal uses for flywheels: To create precessional forces (such as gyroscopes) and to store kinetic energy. This program demonstrated the use of optimized energy storage flywheels for the purpose of supplying peak energy demands in aircraft flight control and utility actuation function duty cycles.

Three simulators were used. One was a tilt table for investigation of precessional loads on bearings. Another was the XB-70 landing gear simulator for demonstration of both hydraulic and mechanical power extraction from flywheels in the performance of intermittent type of aircraft actuation duty cycles. A third simulator demonstrated 1,500 hours of aircraft flight control operation, wherein a bank of three XB-70 elevons were supplied one third of their peak hydraulic flow from a motor-pump driven by a flywheel. This test demonstrated 300 flights of 5 hours each.

These endurance test demonstrations were the second phase in a two-phase program. Phase I effort established analytically that very substantial weight savings can be made in aircraft through the use of energy storage substations in secondary power subsystems.

The three simulators were driven by three identical energy storage substations. The only failures that occurred to the substations throughout the testing were associated with rotary seals and support bearings for the flywheels, and motor-pump rotating groups.

The program very successfully demonstrated the operating characteristics and the feasibility of using flywheel energy storage substations in the accomplishment of aircraft actuation functions. The program also verified the validity of the weight saving estimates made in Phase I.

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Section I

GENERAL

This report covers the Phase II activities of a program to demonstrate the overall feasibility, in terms of performance, reliability and weight, of energy storage substations employing a flywheel as the basic energy storage device. The energy storage concept embraces an approach wherein the energy represented by the surplus power available in the secondary power subsystem during a greater portion of a typical air vehicle mission is stored in one or more substations. The substations are located as close as possible to the point of power use.

It had been shown during Phase I studies (refer to AFAPL TR 66-29) that this approach offered the possibility of saving significant amounts of weight. Therefore, the basic function of this Phase II program was to demonstrate the reliability of the flywheel and to more accurately evaluate the potential weight savings. This demonstration program was also intended to show the dynamic response characteristics of the energy storage substation when used in flight control-type systems, and the durability of the device when used in both flight control and utility applications.

As previously indicated, the flywheel is the "heart" of the system. It is the device which stores the energy through increased rotational speed (and increased internal stress) and gives up energy by slowing down. For this reason, before discussing the demonstration program, it would be well to review some of the flywheel characteristics which were used.

1. The flywheel was made from a high endurance limit (140,000 psi) stress steel which was determined in Phase I to be the most satisfactory material currently available.

2. The cross section of the flywheel was of the so-called "optimized disc" shape. The sides of the disc were double reversed exponential curves thicker at the hub than at the tip. This cross section approached as closely as possible the ultimate, but unattainable, objective of uniform stress throughout the material.

3. At a given stress level, the "optimized disc" configuration has the same specific energy storage (i.e., energy stored per pound of weight) irrespective of diameter. This means that a large diameter, thin, and slow-turning flywheel of the "optimized disc" configuration will store the same energy at the same weight as a small diameter "chunky" flywheel turning at high rotational speed.

4. In addition to its energy-storage characteristics, the flywheel is a gyroscope. This means that when mounted on a maneuvering platform (such as an air vehicle) which maneuvers in three planes, the flywheel will generate significant and sometimes destructive precessional loads in its support bearings.

5. All other things being equal, the precessional loads are a function of flywheel diameter and bearing support spacing. The smaller the diameter and the wider the support spacing the lower the precessional loads. Therefore, to reduce precessional loads, a flywheel should be of as small a diameter and as "chunky" a cross section as possible.

6. Windage losses are a major determinant of flywheel energy storage effectiveness. All other things being equal, flywheel windage losses are a function of the density of the gas surrounding the flywheel and the diameter of the flywheel. The less dense the gas and the smaller the flywheel diameter the less the losses.

As a basis for setting up the demonstration tests, typical aircraft actuation functions have been divided into two general classifications which are identified as "intermittent" and "continuous." Functions fitting the "intermittent" classification are defined as those in which short periods of peak power demand are followed by long quiescent periods of zero power demand. The speed brake and flap systems of most airplanes fit this description as do the landing gear and rocket package extension systems. On the other hand the "continuous" classification is defined as one in which a continuous low level of power demand exists (usually less than 10 percent rated) upon which is superimposed intermittent periods of high power demand (approaching 100 percent rated). Functions fitting this classification are frequently designated "flight control" type and are exemplified by the operation of ailerons, elevons, rudders, air inlet control systems, etc. It then follows that the duty cycles associated with these two functional classifications establish design criteria sufficiently diverse that separate demonstrations were warranted.

For the purposes of this program, the main landing gear extend and retract functions and the flight control elevon actuation function of the XB-70 aircraft were selected as best exemplifying the "intermittent" and "continuous" applications respectively for the endurance test demonstrations. Three simulators were used in the accomplishment of these objectives. These are:

1. A Tilt Table

2. A Flight Control Simulator

3. A Landing Gear Simulator

Each of the above units is described subsequently in detail; however, it is significant to point out that due to careful selection of functions to be demonstrated and tests to be run, it was possible to use essentially the same type of energy storage substation assembly for each of the three demonstrations.

The planned program of testing was as follows:

Tilt Table Testing	100 hours
Flight Control Testing	1,500 hours

Landing Gear (hydraulic power extraction)	10,000 cycles
Landing Gear (mechanical power extraction)	10,000 cycles

Tilt table testing was essentially completed prior to starting either the intermittent or continuous duty cycle endurance testing. These first tests established several important facts and cleared up many areas where only projected thinking was available, on which to base operational procedures for the simulators.

Section II

ENERGY STORAGE SUBSTATION

The basic function of the energy storage substation demonstrated during this program was to store the surplus energy available in an aircraft hydraulic system during periods of steady-state (low) demand and make it available for use by the power using (output) device during periods of peak demand. Of the four demonstration tests conducted, three extracted power from the hydraulic system for storage purposes and returned it to the hydraulic system on demand. The fourth (i.e., the mechanical intermittent duty test) also extracted the power for storage from the hydraulic system but delivered the demand power mechanically in the form of rotating shaft power.

In the interests of simplicity the substation designed for this program was set up so that, with only slight modifications, it could be adapted to any of the four demonstrations. The resulting substation is shown in figure 1 and the photograph of figure 2. All three units built were essentially identical except for interchanging different motor-pump units. Figure 1 shows a New York Air Brake motor-pump installed whereas figure 2 shows a Vickers motor.

As can be seen in figure 1 the energy storage substation consists of a flywheel (to store energy), a hydraulic motor-pump (acting as a motor to keep the flywheel wound up and acting as a pump to extract energy on demand), a gearbox (to couple the motor-pump to the high-speed flywheel and to provide a power takeoff pod for the extraction of mechanical power), and a service pump group (to lubricate and scavenge the substation and evacuate the flywheel case). Each of these components is discussed in detail in the following paragraphs.

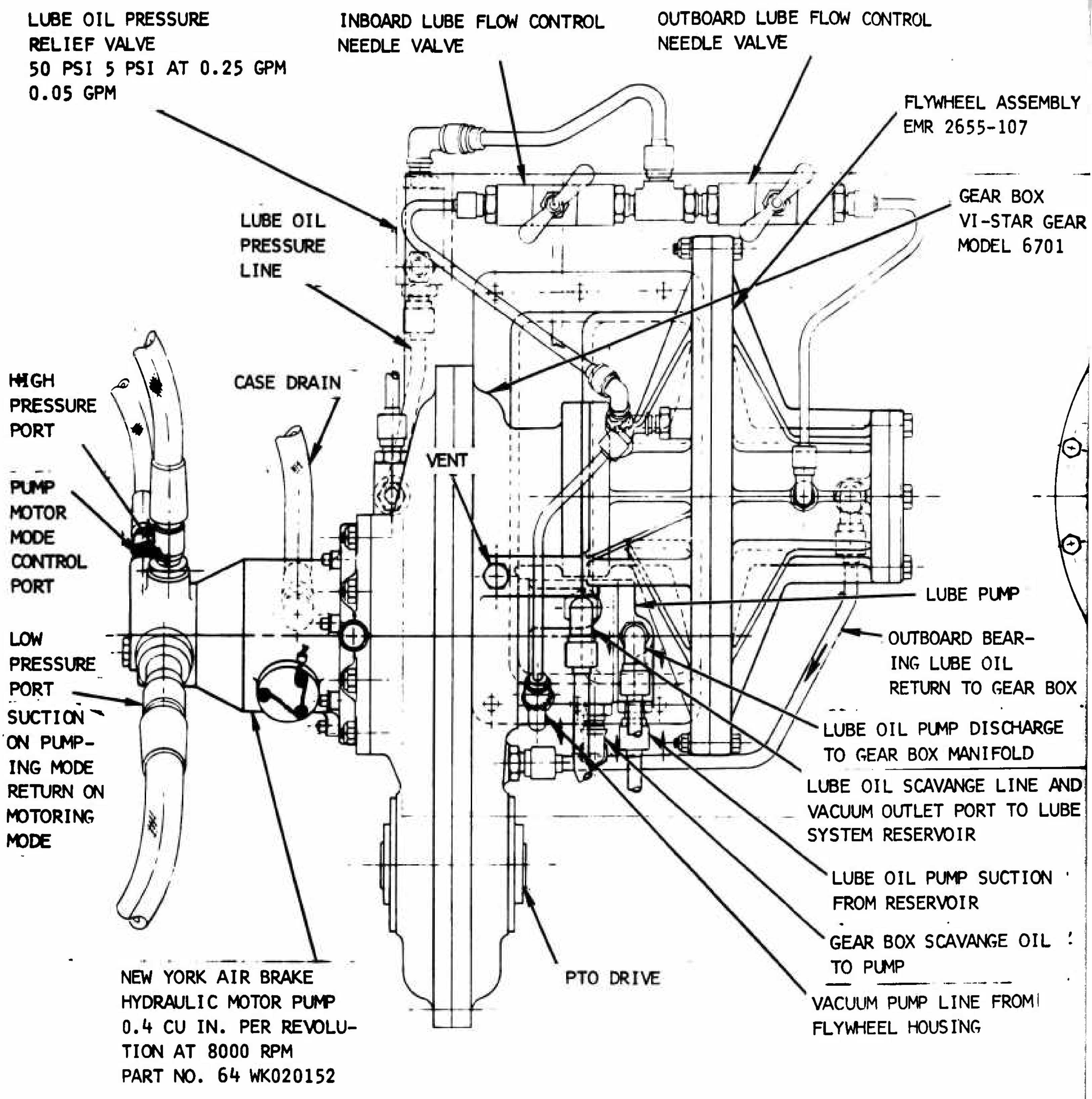
A. FLYWHEEL DESCRIPTION

The flywheels for all of the energy storage substations were purposely designed the same, even though optimization would have caused them to each be of different capacities. Identical flywheels were chosen in the interest of overall program effectiveness. First, they would be essentially interchangeable on the simulators; and second, a proper evaluation of their true capabilities could be more easily made if disruptive factors resulting from size and configuration differences did not enter into the analysis of data.

Table I is a tabulation of the flywheel geometry and characteristics. It was created by using direct ratios of the flywheels computed and tabulated during the Phase I program.

Total kinetic energy was determined from the values of required useful kinetic energy and speed reduction. The following relationship was used:

$$KE_{Total} = KE_{Useful} \frac{(\text{Maximum Speed})^2}{(\text{Maximum Speed})^2 - (\text{Minimum Speed})^2}$$


LUBE OIL PRESSURE RELIEF VALVE 50 PSI 5 PSI AT 0.25 GPM 0.05 GPM
INBOARD LUBE FLOW CONTROL NEEDLE VALVE
OUTBOARD LUBE FLOW CONTROL NEEDLE VALVE
FLYWHEEL ASSEMBLY EMR 2655-107
GEAR BOX VI-STAR GEAR MODEL 6701
LUBE OIL PRESSURE LINE
HIGH PRESSURE PORT
CASE DRAIN
VENT
PUMP MOTOR MODE CONTROL PORT
LOW PRESSURE PORT
SUCTION ON PUMPING MODE RETURN ON MOTORING MODE
LUBE PUMP
OUTBOARD BEARING LUBE OIL RETURN TO GEAR BOX
LUBE OIL PUMP DISCHARGE TO GEAR BOX MANIFOLD
LUBE OIL SCAVANGE LINE AND VACUUM OUTLET PORT TO LUBE SYSTEM RESERVOIR
LUBE OIL PUMP SUCTION FROM RESERVOIR
GEAR BOX SCAVANGE OIL TO PUMP
VACUUM PUMP LINE FROM FLYWHEEL HOUSING
PTO DRIVE
NEW YORK AIR BRAKE HYDRAULIC MOTOR PUMP 0.4 CU IN. PER REVOLUTION AT 8000 RPM PART NO. 64 WK020152

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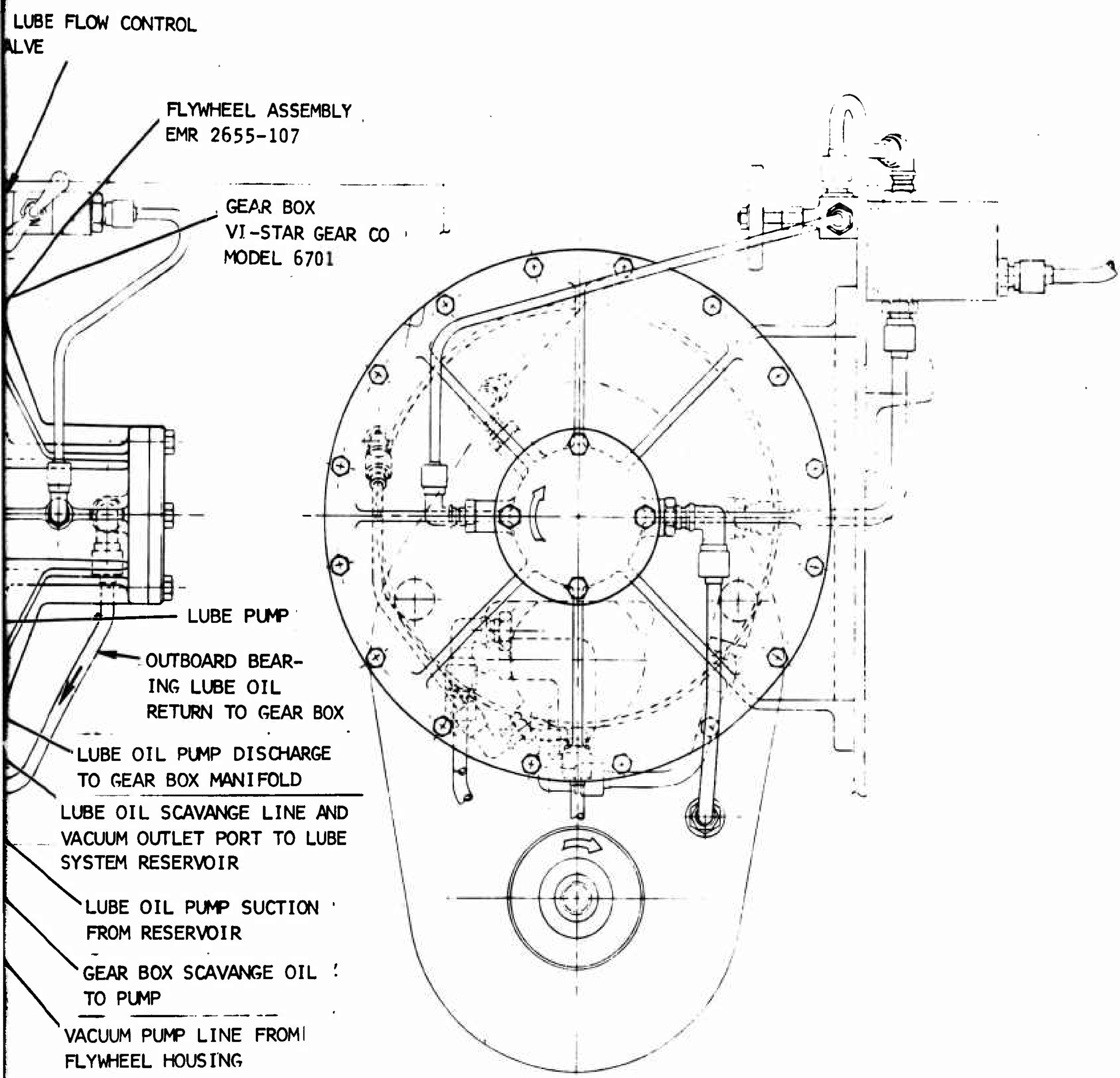

Figure 1. Energy Storage Substation

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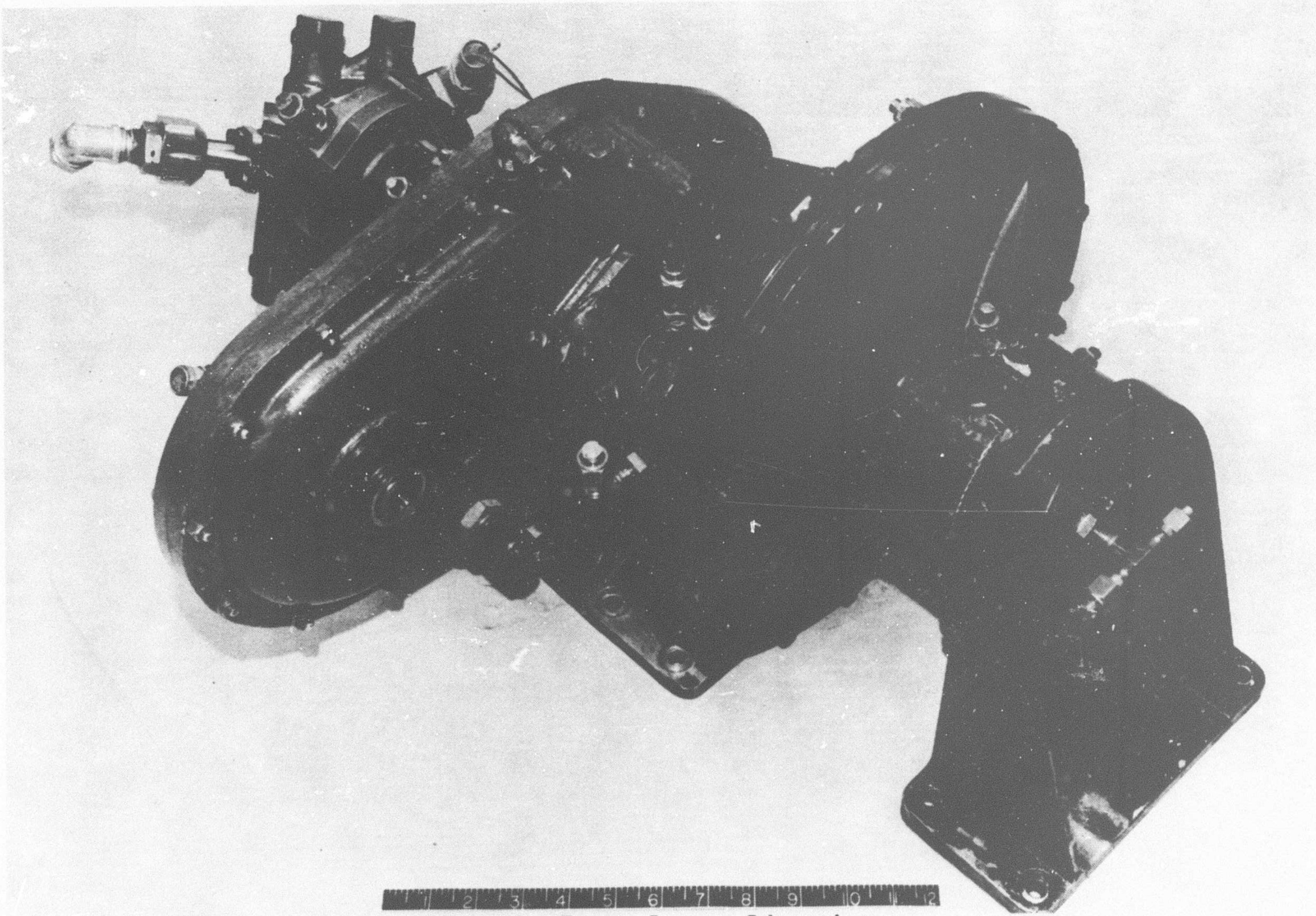

Figure 2. Energy Storage Substation

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Table I

FLYWHEEL GEOMETRY FOR AN OPTIMIZED DISK (APPROXIMATELY CONSTANT STRESS)
(MATERIAL - STEEL)

Outside Radius (R)	= 5.000 inches	Design Stress (S)	= 97,222 lb/sq in.
Poissons Ratio (MU)	= 0.300	Density (RHO)	= 0.285 lb/cu in.
Kinetic Energy (KE)	= 4,000,000 in.-lb	Weight (WT)	= 14.058 lb
Angular Velocity (OMEGA)	= 5,487 rad/sec	KE/WT	= 284,538 in.-lb/lb
Rad of Gyration (K)	= 2.702 inch	Moment of Inertia (I)	= 0.265 lb-in.-sq sec

Position	Radius	Thickness (T)	T/2	S (Tangential)	S (Radial)
1	5.000	.113	.056	69,240	0
2	4.750	.148	.074	76,355	23,692
3	4.500	.192	.096	81,519	40,901
4	4.250	.246	.123	85,311	53,591
5	4.000	.311	.155	88,126	62,926
6	3.750	.386	.193	90,238	69,966
7	3.500	.473	.236	91,838	75,300
8	3.250	.572	.286	93,062	79,379
9	3.000	.681	.340	94,006	82,526
10	2.750	.801	.400	94,740	84,971
11	2.500	.928	.464	95,313	86,883
12	2.250	1.061	.530	95,765	88,388
13	2.000	1.195	.598	96,120	89,573
14	1.750	1.328	.664	96,401	90,508
15	1.500	1.454	.727	96,621	91,242
16	1.250	1.572	.786	96,792	91,813
17	1.000	1.674	.837	96,993	92,249
18	0.750	1.758	.879	97,019	92,569
19	0.500	1.822	.911	97,085	92,788
20	0.250	1.860	.930	96,123	92,916
21	-0.000	1.874	.937	97,136	92,958

The useful kinetic energy was established from the duty cycle required of the simulators, and the speed reduction was established by determining the minimum speed that could be tolerated by the pump and still meet the flow requirements.

Having established a total kinetic energy requirement from the foregoing it was then necessary through sizing a number of flywheels to make a weight and trade-off study of windage losses, precessional loads resulting from bearing spacing, housing weight, housing design, and general space and mounting provisions.

As a result of the above analysis, it was found that a 12-inch diameter flywheel was the most efficient in terms of energy storage versus weight at 52,400 rpm when considering the flywheel alone. However, this was not necessarily true when considering the overall flywheel assembly consisting of the flywheel, its housing, seals, and bearings. The analysis indicated that the large OD and relatively slender cross section of the 12-inch diameter flywheel when compared to a lesser diameter "chunkier" flywheel led to disproportionately large windage losses and placed greater demands on the bearings to resist precessional loads (refer to item 5, page 2). At the same time, the "slender" flywheel with a given bearing support spacing has a lesser inherent shaft stiffness. The weight saved in the slender flywheel was more than offset by increased weight in the housing and bearings; therefore, the present 10-inch diameter flywheel was selected as most nearly approaching optimum for the conditions. The 10-inch diameter selected could have been operated at 6,584 radians/second without exceeding 140,000 psi tensile stress; however, in consideration of the interrelated and sometimes conflicting requirements for specific energy storage, windage loss, and bearing life, operation at 5,487 radians/second (i.e. 52,400 rpm or 97,222 psi tensile stress) was selected.

The flywheel was mounted in ball bearings with features generally as follows:

25 mm bore (.9843 in.)
2.0472 outside diameter
17 mm width
11/32 dia ball (11 each bearing) single ball row, with angular
  contact and outer race riding cage, light series

The flywheel shroud was contoured to fit the flywheel exponential shape with a uniform clearance of 1/8 inch, thus reducing aerodynamic losses. To further reduce aerodynamic losses, the shroud was evacuated. Figure 3 is a plot of the horsepower losses versus vacuum. Rotary carbon seals were used between the evacuated flywheel and its bearings.

A cross-sectional drawing of the flywheel in its housing is shown in figure 4. Figure 5 is a photograph of one of the disassembled units after 34.5 hours of testing.

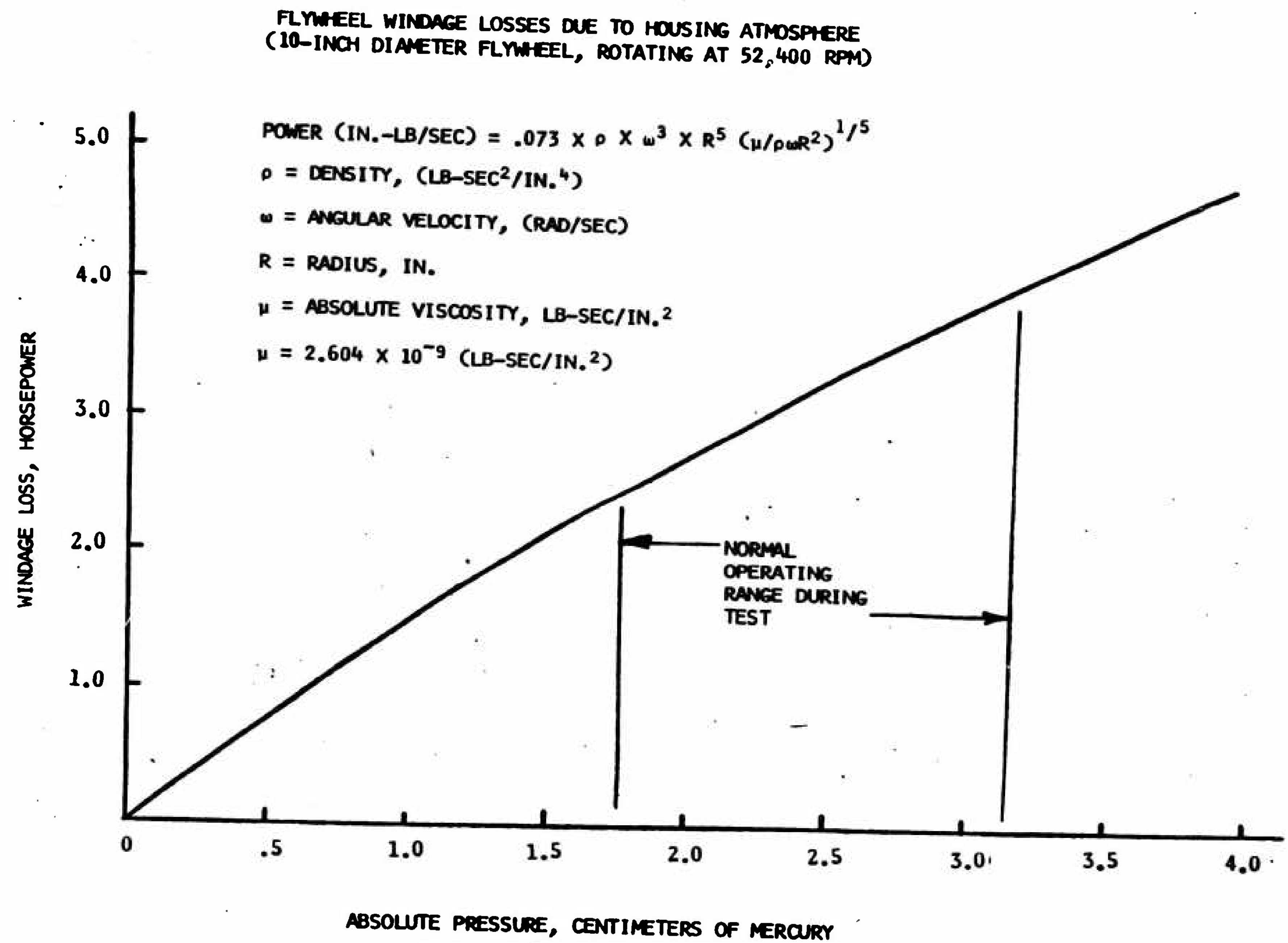


Figure 3. Aerodynamic Losses

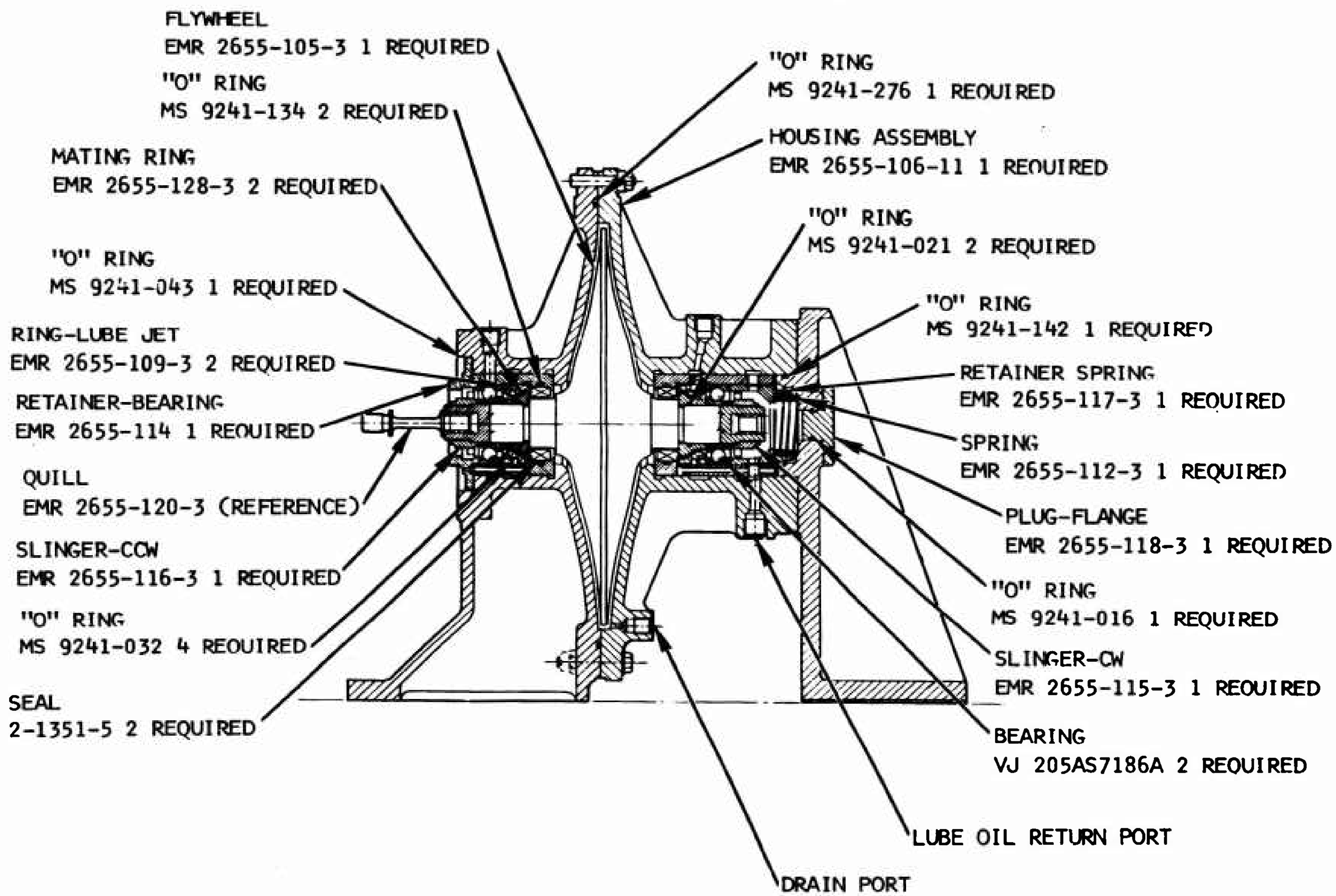


Figure 4. Flywheel Assembly

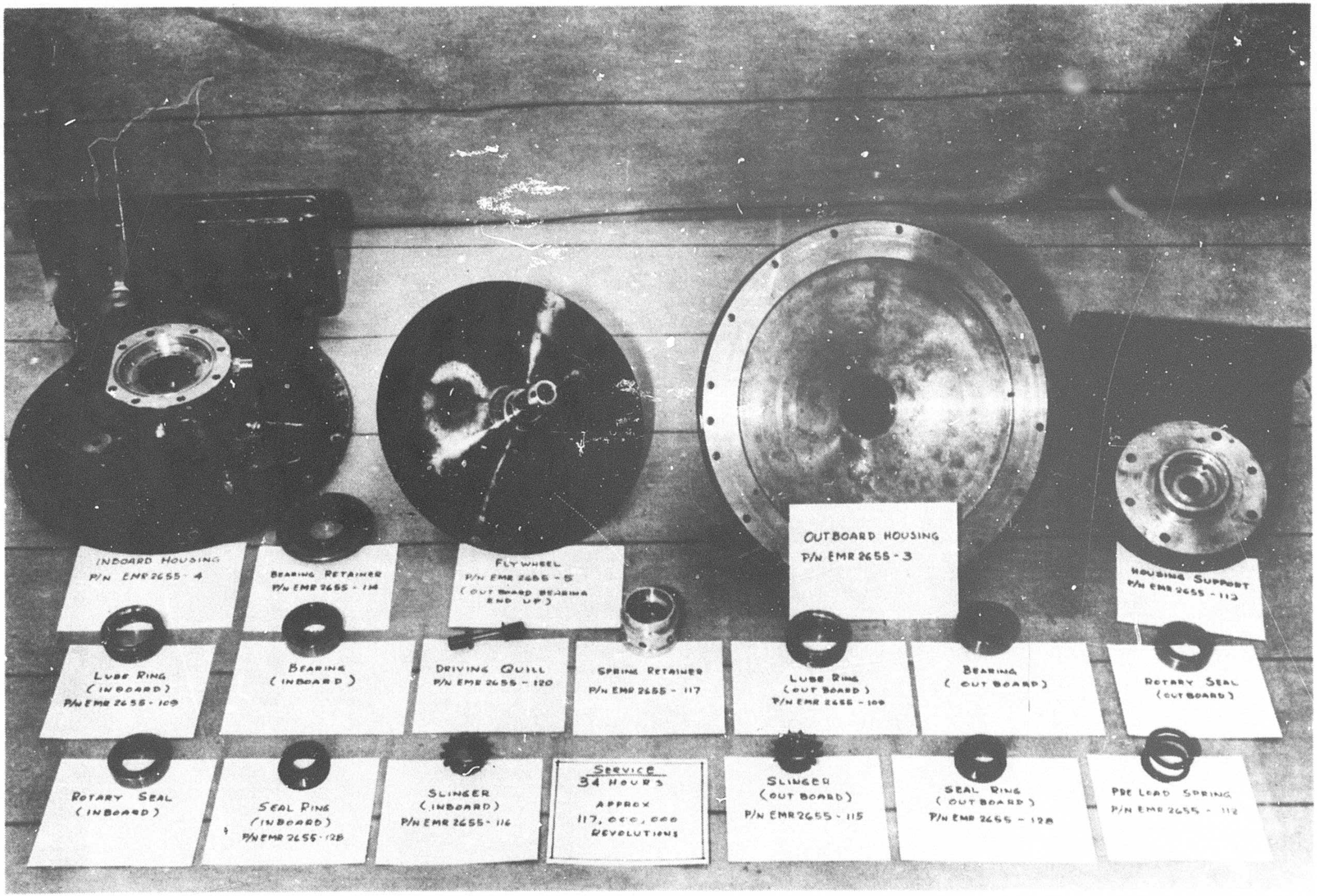


Figure 5. Flywheel Assembly

B. GEARBOX DESCRIPTION

The gearbox consists of a housing which contains three gears and mounts a gear-type service pump group. Figure 6 is the assembly drawing used for its manufacture and table II is an itemized parts list keyed to the callouts of figure 6.

The gearbox housing is fabricated from a heavy sectioned-aluminum sand casting. It is split down the middle and the two halves are doweled together so that all the gear shaft bores and bearing seats were machined in one setup to the extreme accuracy necessary for high-speed operation. The gears are hardened and lapped to attain the tooth profile accuracy required.

The larger gear has 190 teeth and is axially aligned with motor-pump mounting provisions which are on one of the housing halves. The motor-pump drives this gear through a spline at 8,000 rpm. This gear meshes with two other gears. One of these gears has 29 teeth, and therefore rotates at 52,414 rpm for introducing power into, and extracting power from, the flywheel through a splined quill shaft. The flywheel housing bolts rigidly to the gearbox and is machined to maintain accurate alignment between this gear and the flywheel. The gearbox is supported by this attachment which consists of a flange and two "outrigger" pads. This gives a rigid tripod-type support.

The other gear, which is in mesh with the motor-pump driven gear, has 175 teeth and rotates at 8,686 rpm. It is referred to as the PTO shaft gear and drives the mechanism for demonstrating mechanical power extraction in conjunction with the Intermittent Duty Cycle Simulator.

The service pump group is mounted in axial alignment with the motor-pump, on the opposite side (flywheel side) of the gearbox; therefore, it is also driven at 8,000 rpm. It consists of three gear pumps, one for generating lubrication pressure, one for lubricant scavenging from the flywheel bearings and gearbox, and the other for flywheel cavity evacuation. The pumps are located with respect to each other in the above order, with the vacuum pump being nearest to the gearbox.

Pump displacements are as follows:

Lubricant pressure pump	.038 $in.^3$/rev
Lubricant scavenge pump	.075 $in.^3$/rev
Vacuum pump	.325 $in.^3$/rev

Lubricant from the pressure pump is squirted at the gear teeth in two places. This lubricates the gears and at the same time creates a mist which lubricates the bearings. A part of the lubricant flow is piped to the flywheel bearings as shown in the flywheel assembly drawing of figure 4.

C. MOTOR-PUMP DESCRIPTION

The basic motor-pump selected for this program was manufactured by the New York Airbrake Company, and has automatic pressure-sensing control for determining whether it should be motoring at full displacement or should be in the pumping mode. In the pumping mode, displacement is variable and is

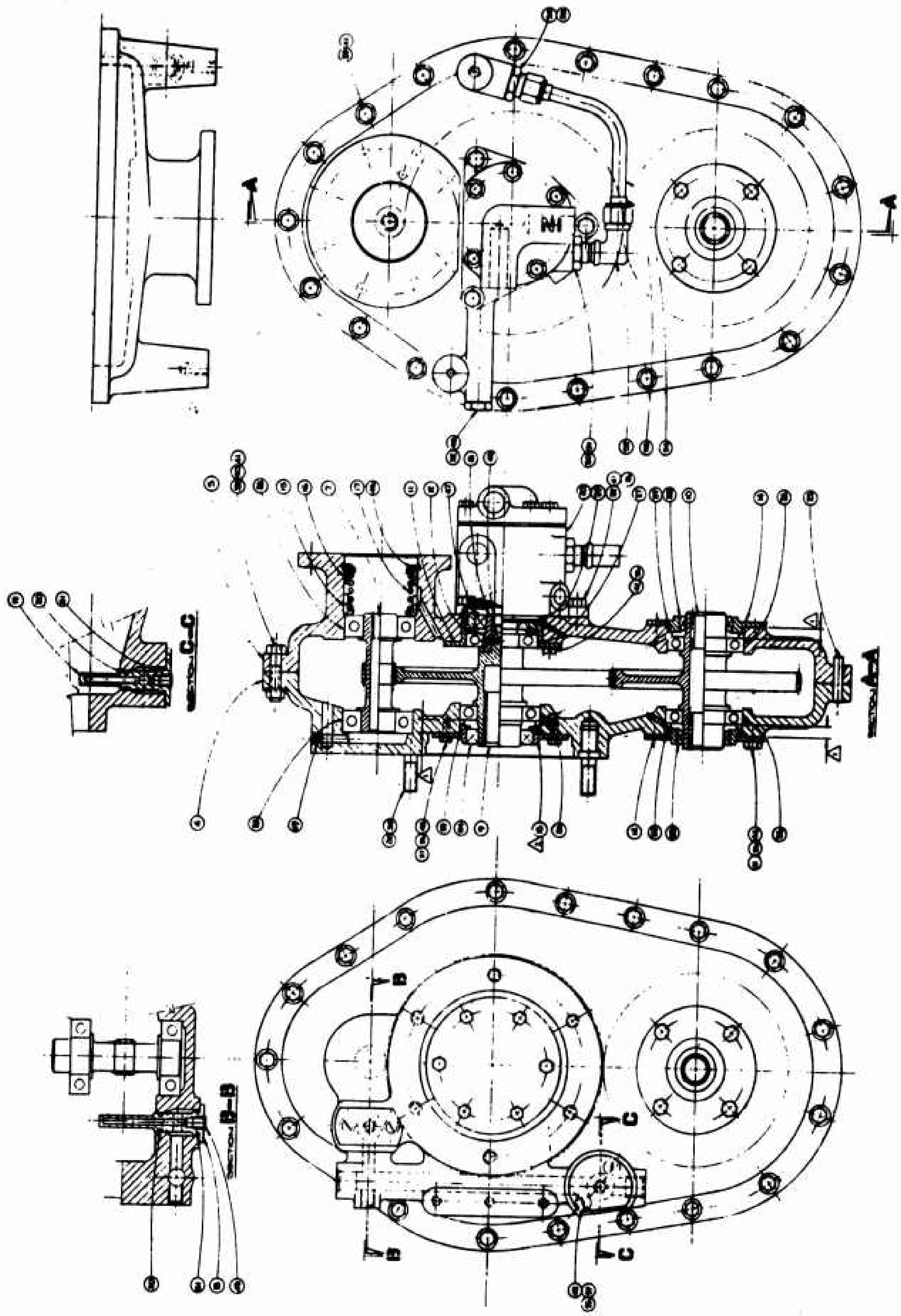


Figure 6. Gearbox

Table II

GEARBOX ITEMIZED PARTS LIST

Item Number	Nomenclature
4,5	Housing - input
7	Pinion
8	Race
9, 10	Gear
11	Sleeve
12	Spacer
13, 14	Retainer
15, 16	Guide
17	Spring
18, 19	Oil jet
20	Pump
25	Taper pin
26, 27, 28	Bearing
29, 30, 31, 32, 50	O-ring
33, 34	Seal
35	Stud
36	Lock ring
37, 38, 39, 43, 44	Bolt
40	Nut
41, 57	Washer
42	Screw
45	Pin
46	Retaining ring
47	Clutch
48, 49	Plug
51	"B" nut
52	Elbow
53	Sleeve
54	Tube
56	Lock wire

controlled by pressure. Three of these units were used in the program and were designated "automatic units." All three automatic units have flow regulators built into their control heads.

The Tilt Table Simulator used a modification of the above motor-pump design. The modification allowed manual control of the motoring and pumping mode by means of pressure supplied to an external port in the control head. An externally-attached flow regulator was used to limit motoring speed. This modified "manual" variety of motor-pump was actually a slightly revised version of an existing production unit which could be delivered on an early schedule. This was the principal reason that it was introduced into the program. Figure 7 is a cross-sectional view of this motor-pump.

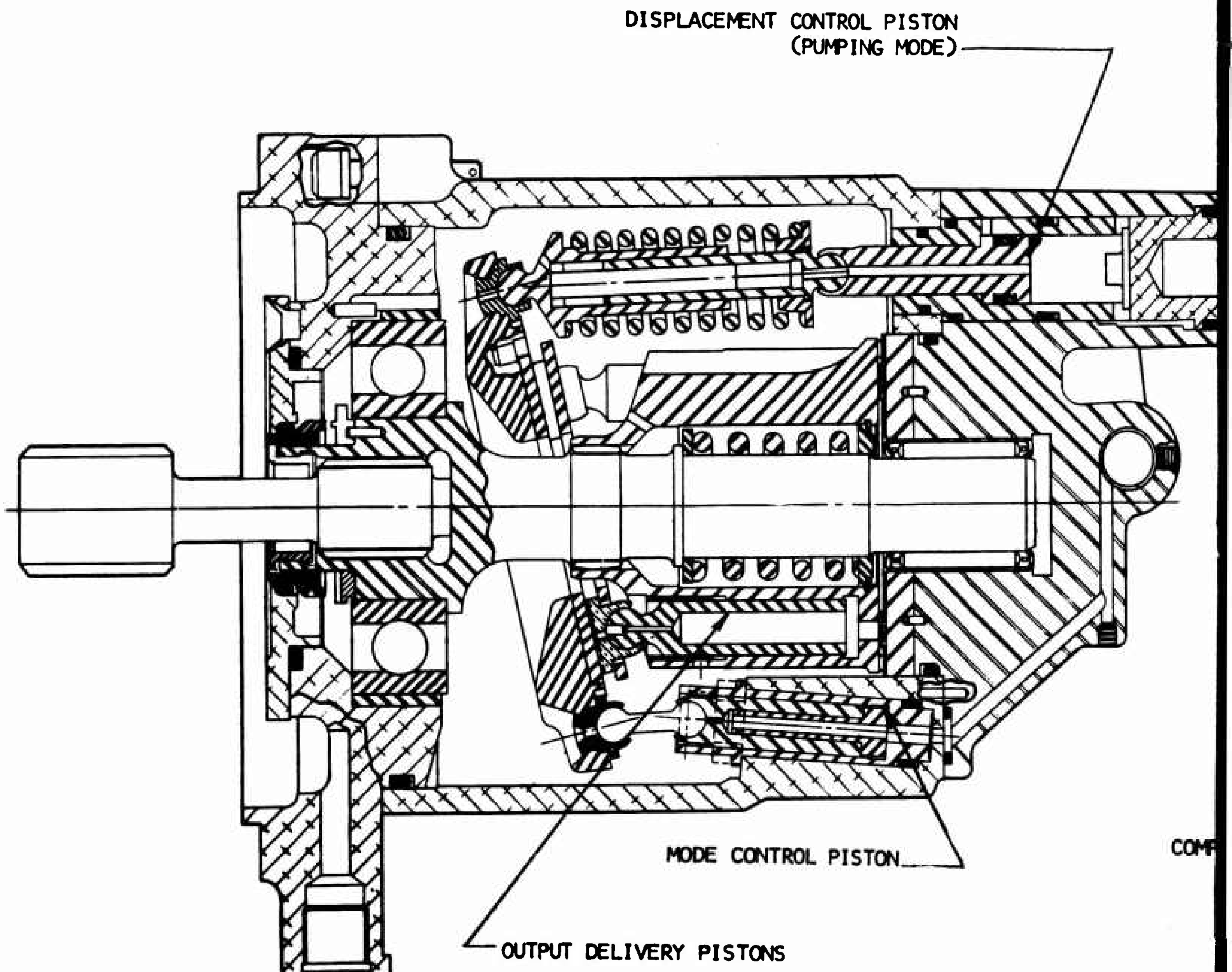
A third variety of motor-pump was used in the Tilt Table Simulator testing. It was manufactured by Vickers, Inc, and was obtained (one unit) from the XB-70 program. It was introduced into the flywheel program, because several early failures of the prototype motor-pump, due to improper operating conditions, occurred and would otherwise have delayed flywheel testing.

A more detailed description for each of the units follows.

Figure 8 is an envelope drawing of the motor-pump with "automatic" pressure sensing. It has a maximum displacement of .4 in.^{3}/rev in both the motoring and pumping modes of operation. The rotating group is the same as that shown in figure 7. Each type contains nine pistons and has an operational speed of 8,000 rpm.

The pressure control of the "automatic" unit positions between full displacement and zero displacement in the pumping mode by regulating a variable pressure level to the displacement control actuator. This is dependent upon the existing aircraft system pressure. A low system pressure causes increased flow from the unit, thus extracting energy from the flywheel for high demands in the aircraft control system duty cycle. A higher aircraft system pressure causes the pumping mode to go to zero displacement. A further increase in the aircraft system pressure causes the mode control valve to port pressure into a chamber of the pressure control unit which moves an additional valve slide. Movement of this valve slide causes the mode control actuator to go to full displacement as a motor. Variations in the aircraft system pressure, above the minimum pressure which puts the unit in motoring mode, do not vary the displacement as a variable motor. The maximum speed is limited by a flow regulator in the high pressure port. The reason that the motor was made to operate as a fixed displacement unit, at all times while in the motoring mode, was to make it possible to limit the speed by means of a flow regulator. In some respects it would have been advantageous to use the motor as a variable displacement unit. For example, it would have smoothed the transition from pumping to motoring. However, to do so would have necessitated the use of some type of shaft speed sensing governor and would have complicated the development and procurement of the motor-pump unit considerably.

The above description also applies to the modified "manual" motor-pump except for the previously mentioned differences in the pressure control valving. Figures 9, 10, and 11 depict the variation. This modified unit requires



DISPLACEMENT CONTROL PISTON
(PUMPING MODE)
MODE CONTROL PISTON
OUTPUT DELIVERY PISTONS

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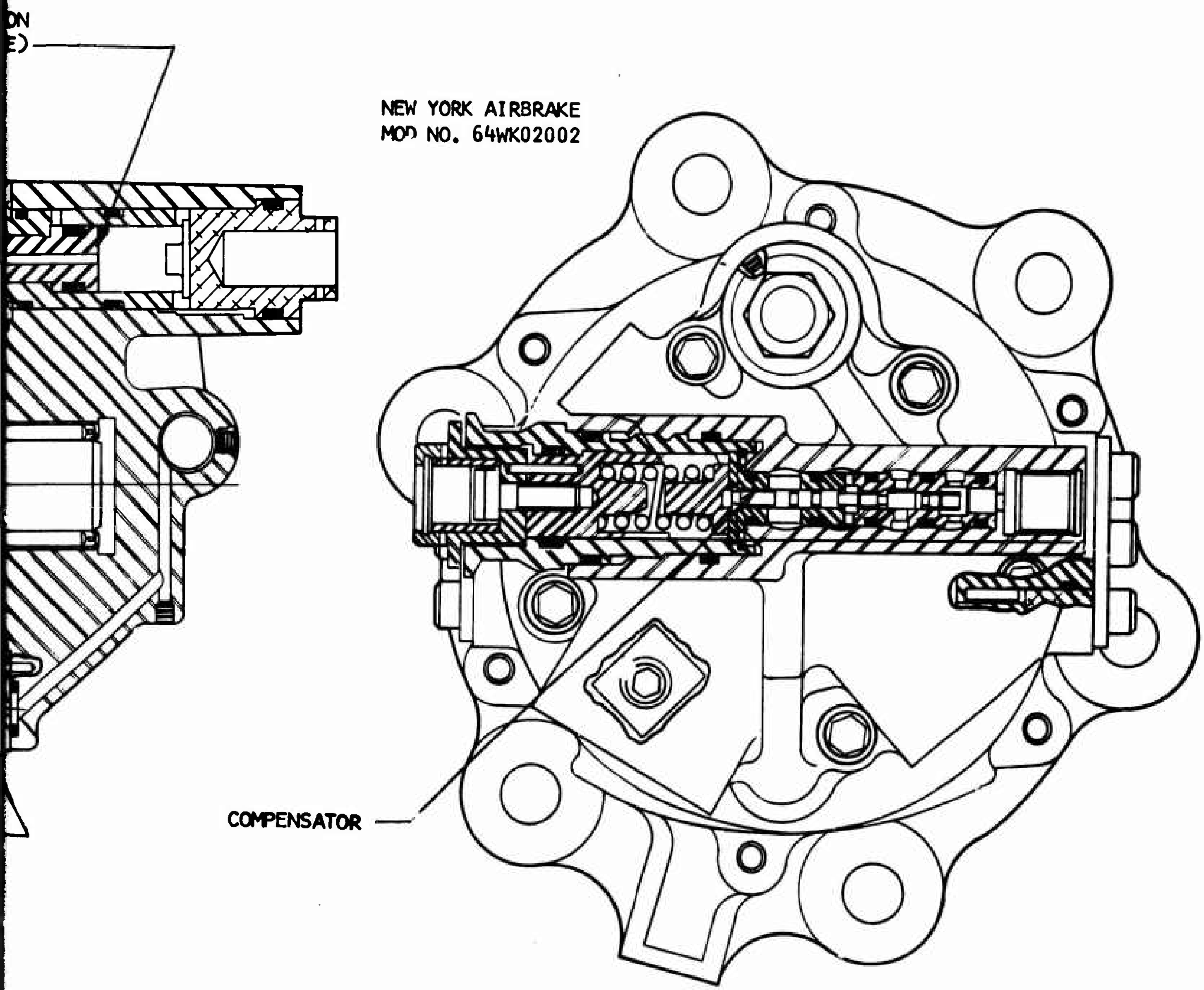

Figure 7. Manual Motor-Pump

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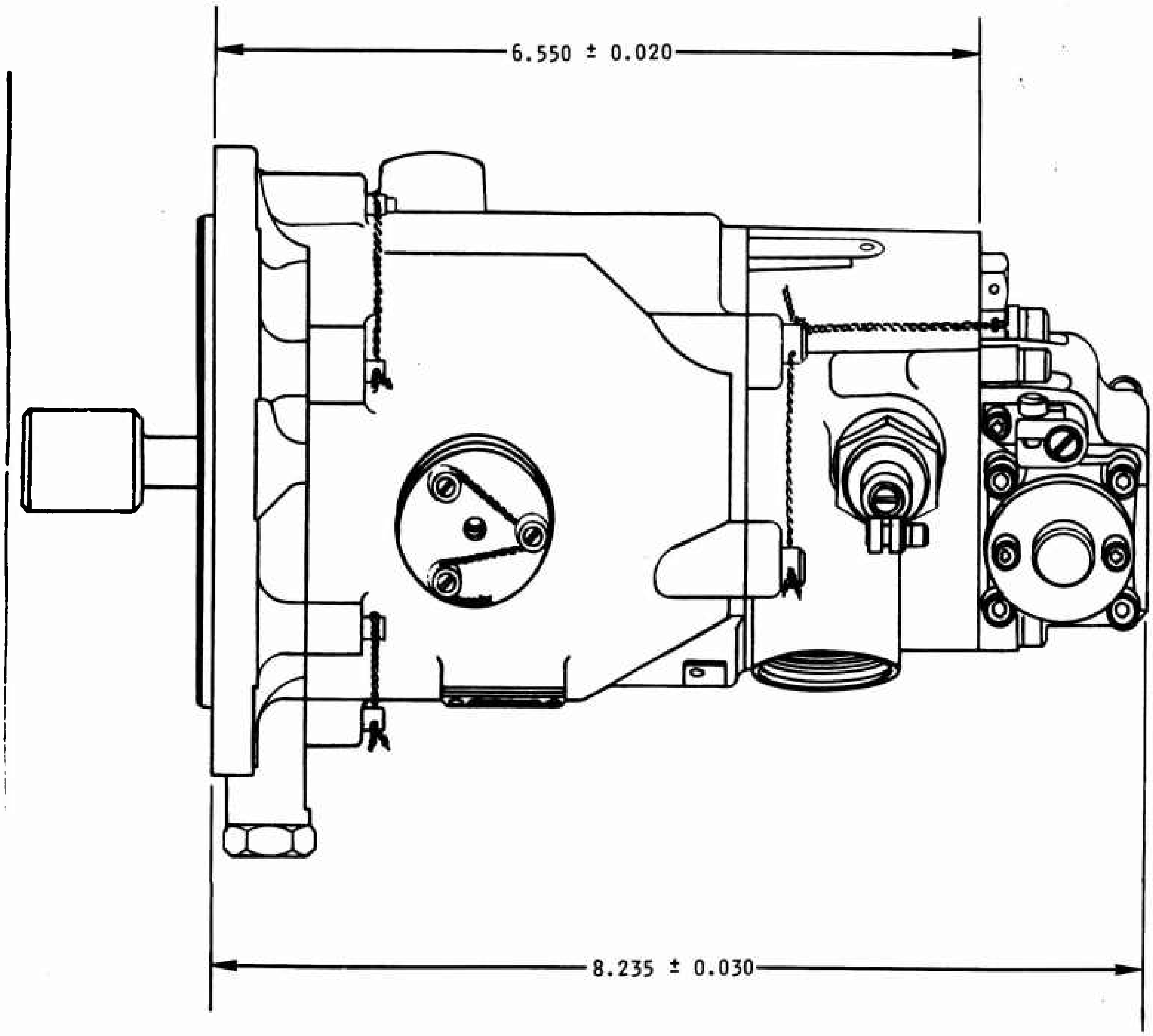

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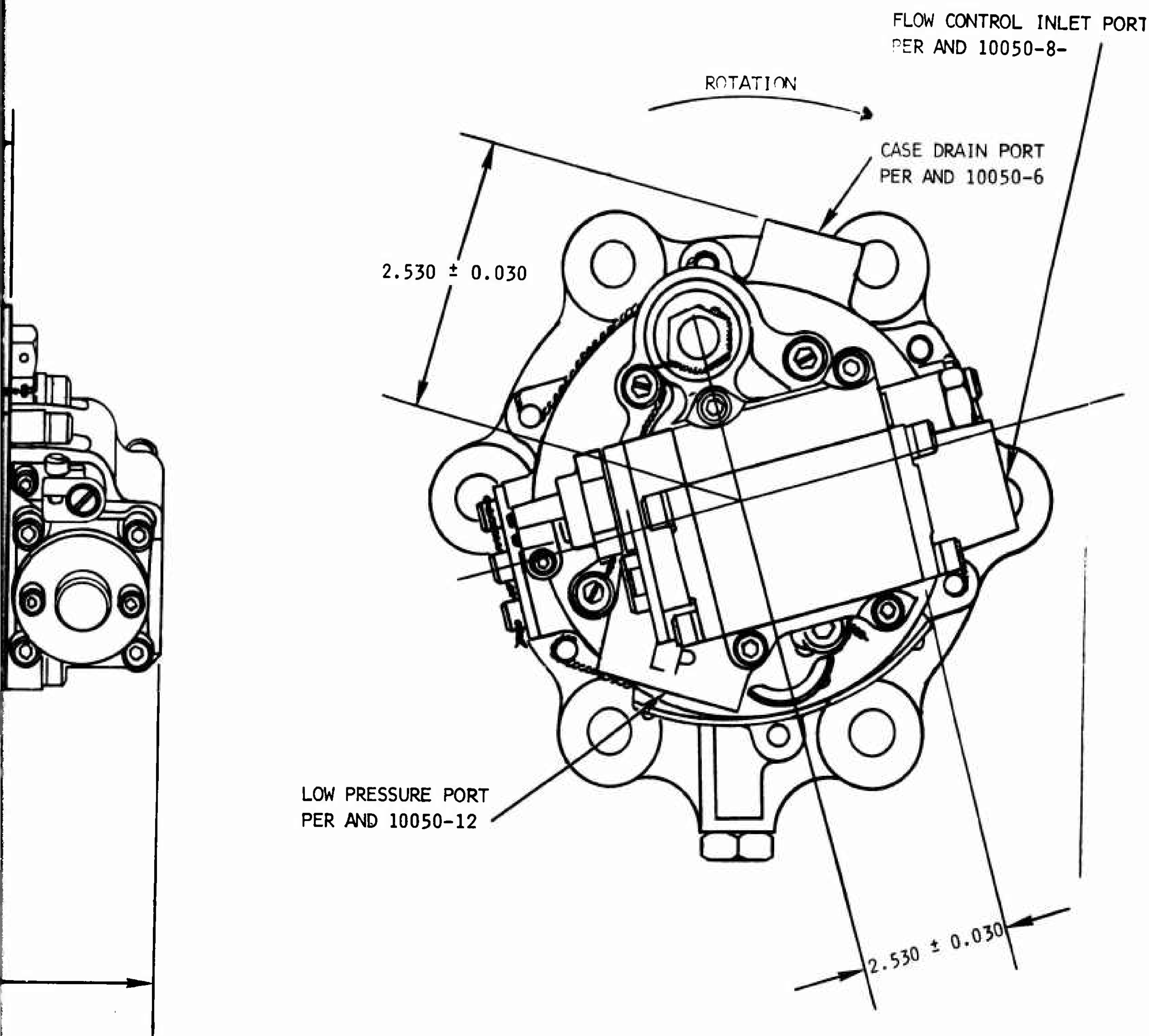

Figure 8. Automatic Motor Pump

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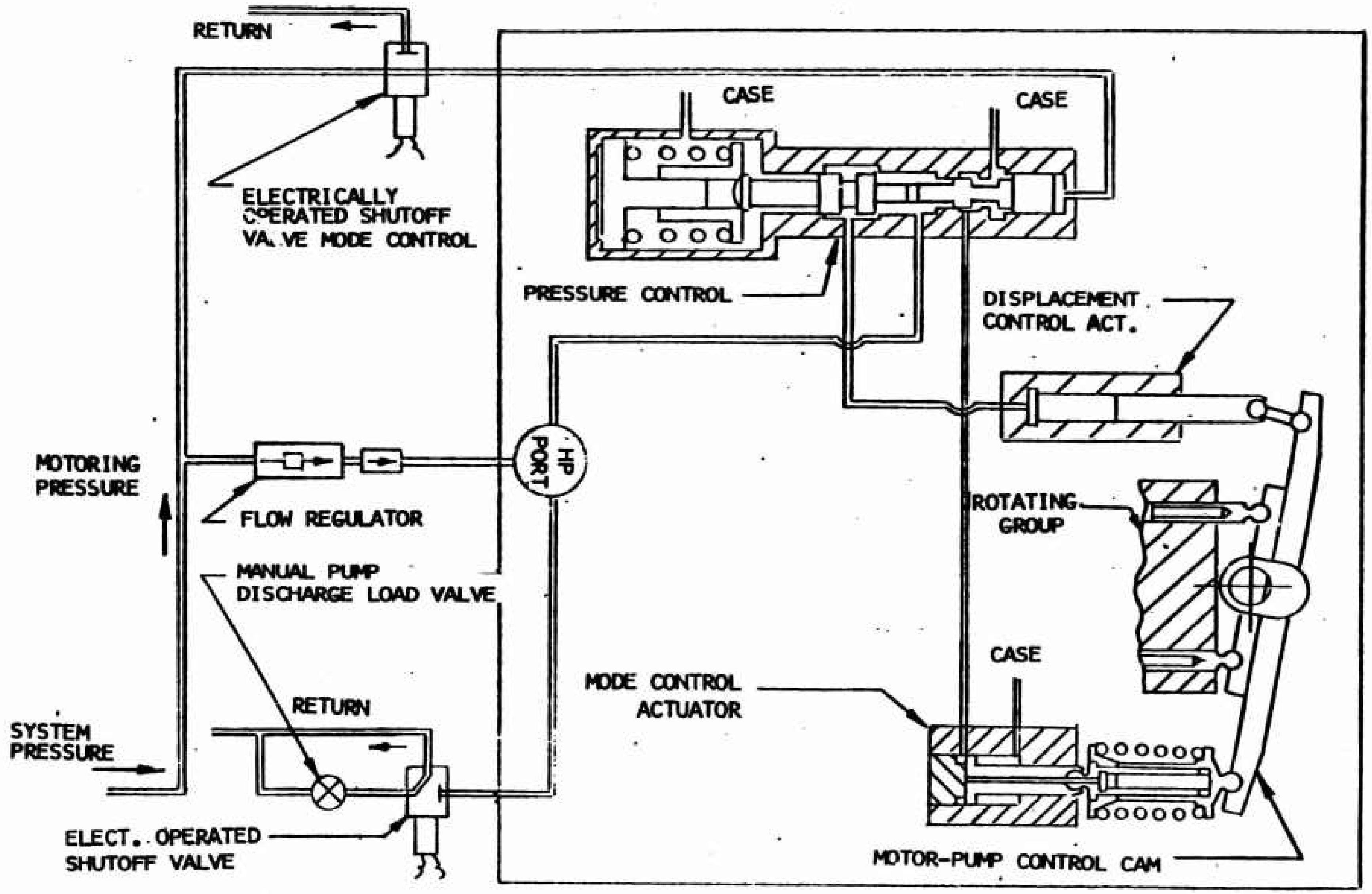


MOTORING MODE

Figure 9. Prototype (Manual) Motor-Pump Configuration

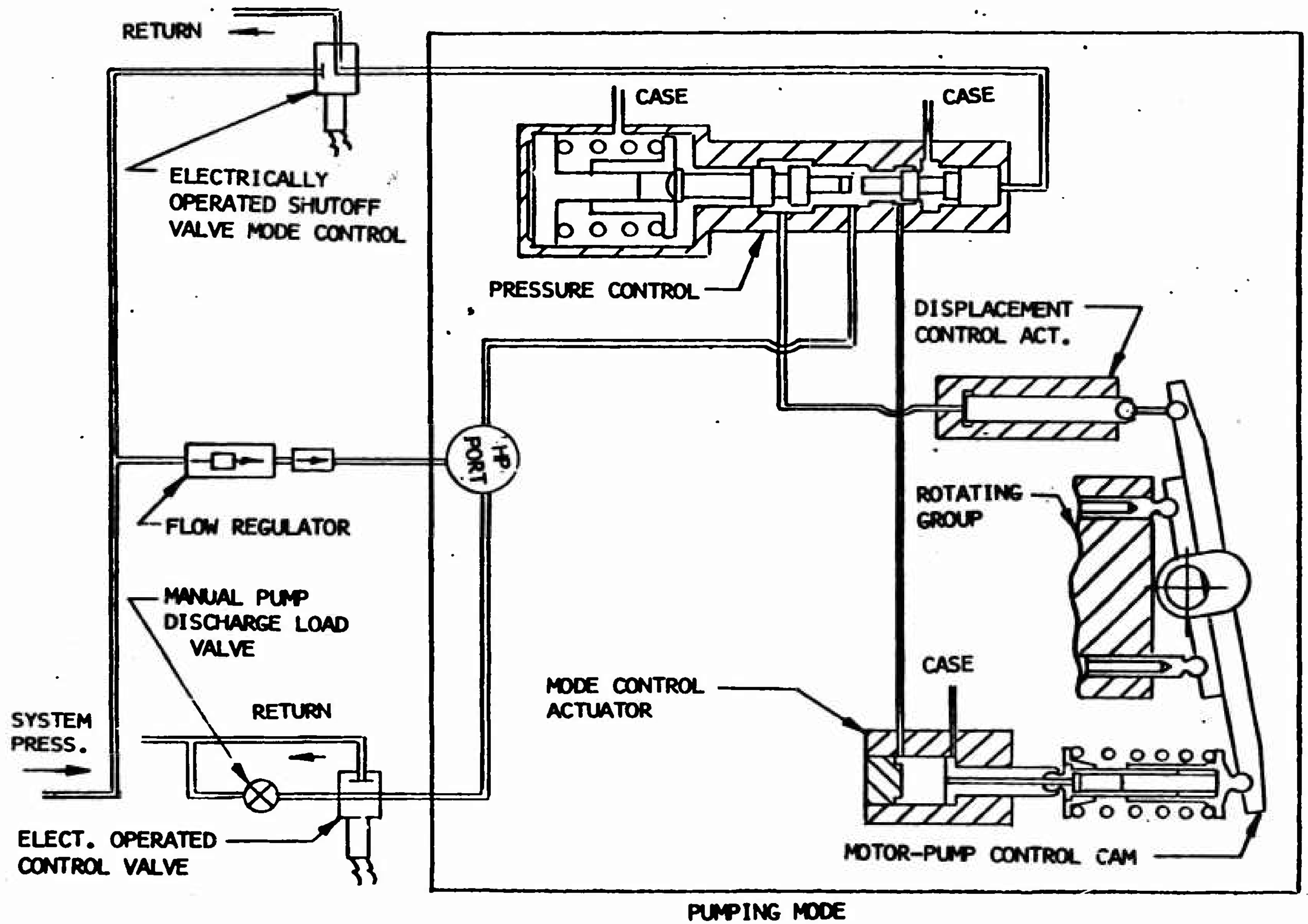


Figure 10. Prototype (Manual) Motor-Pump Control Configuration

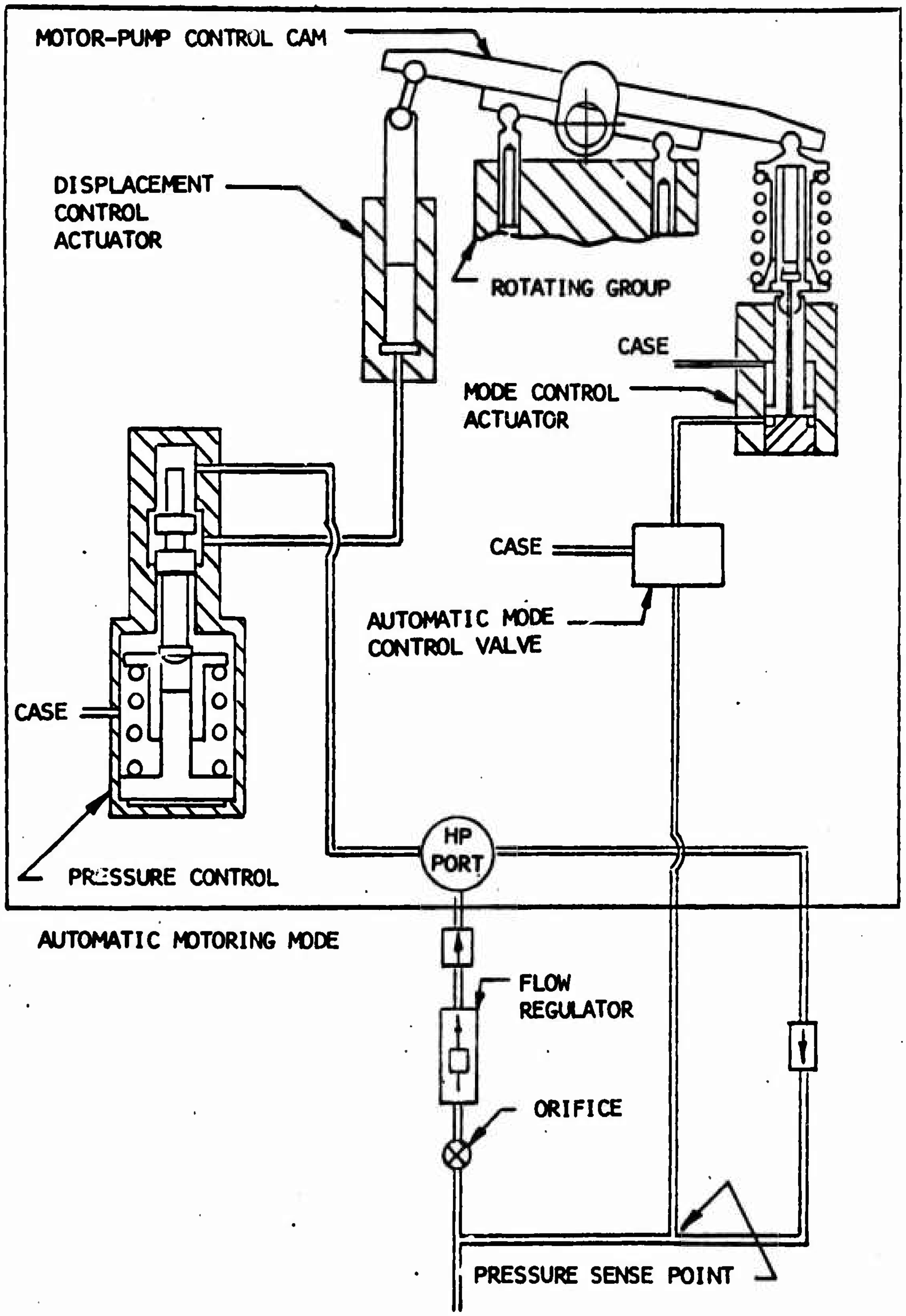


Figure 11. Automatic Motor-Pump Control Configuration

external valving in the test setup to apply dump pressure to the mode control actuator. The shift from motoring to pumping is manually controlled.

Figure 12 is a performance envelope for these New York Airbrake motor-pumps.

The third variety of motor-pump used in the program was from an XB-70 emergency generator application. The rotating group had been developed specifically for the XB-70 wing fold system and was modified for driving the emergency generator by adding a new control head for accurate speed control. The new unit had more displacement, but fortunately had exactly the desired speed built into its control head.

Figure 13 is a schematic diagram of the unit. Other pertinent data are:

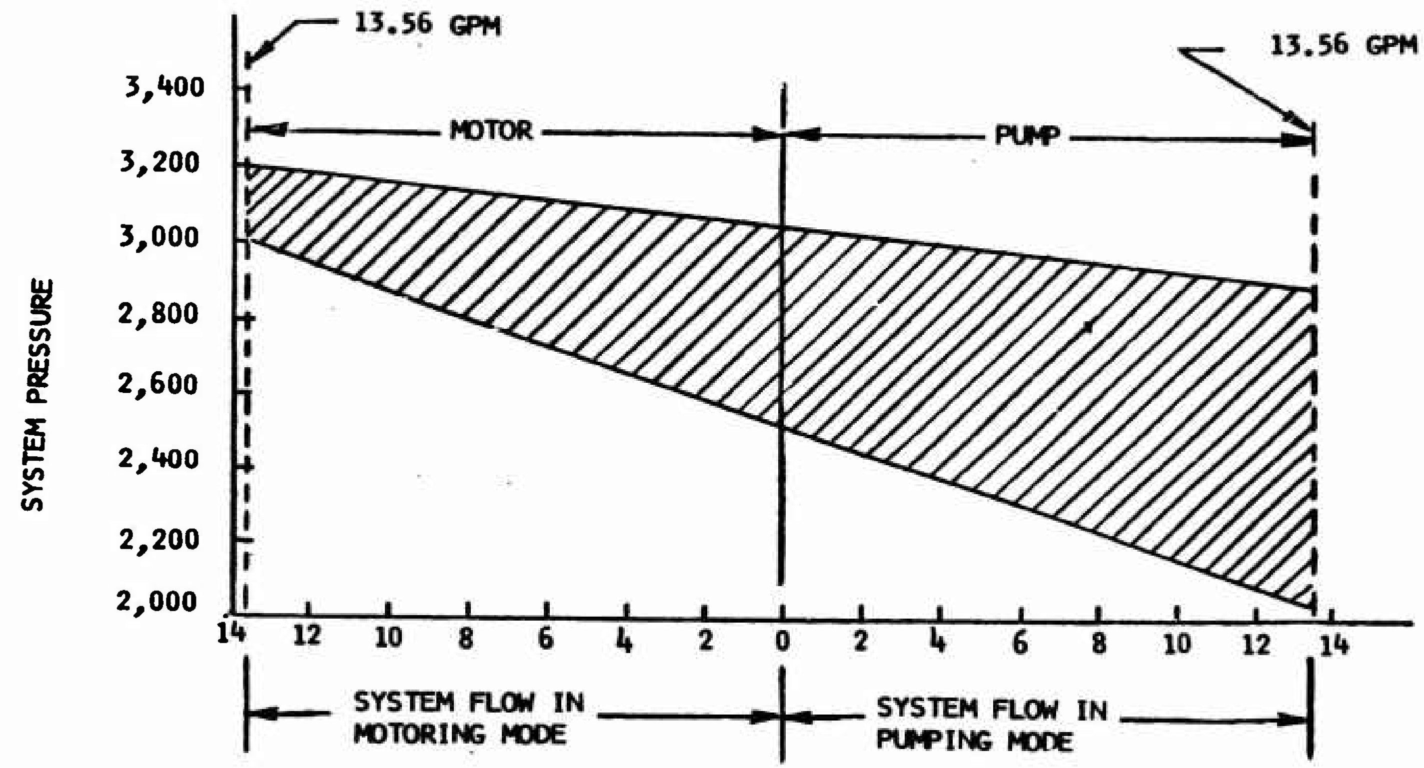
Displacement	.646 $in.^3$/rev
Speed	8,000 rpm
Rated operating pressure	4,000 psi

The application of this unit for the emergency generator required clockwise rotation as viewed from the mounting pad; however, for the energy storage substation application, counterclockwise rotation was required. By rotating the control head 180 degrees it was possible to meet the reversed rotation requirement. This change blocked the case return passage from the return port. To correct this, an external case return line to the reservoir was incorporated. In addition, the tapped mounting holes were reamed to accept the unthreaded shank of the gearbox mounting studs.

As seen in figure 13, this unit incorporates an anticavitation circuit which consists of a fluid recirculation loop. It is designed to protect the rotating group from cavitation when the inlet flow is suddenly reduced while the shaft continues to rotate due to the inertia load of the emergency generator rotor. Although this unit is not a motor-pump, the recirculation loop creates a partial pumping condition which was considered suitable for the application to part of the testing required on the Tilt Table Simulator. A further clarification of this unit's performance will be brought about in covering the actual endurance tests.

D. SUBSTATION WEIGHT

The substation shown in figures 1 and 2 stores 4,000,000 inch-pounds of energy at a weight of 91 pounds. If packaged for an actual air vehicle application, the same unit would weigh less than 60 pounds. The reason for the weight difference is the fact that this unit was standardized, as mentioned previously, so that it could be used in all three test simulators interchangeably. Because of this, certain compromises had to be made. As an example, the heavy feet seen on the flywheel in figure 2 were used to accurately transmit precessional loads to the tilt table bed. An actual air vehicle unit would probably be suspended on a light system of tension struts. Also, the PTO gear portion of the gearbox (seen bulging down in the lower portion of figure 1) is used for only one out of the four tests being conducted (i.e., the intermittent duty cycle mechanical system demonstration). In other words, the gearbox weighs at least one-third more than necessary



NOTE: 8,100 RPM IS MAXIMUM UNDER ALL LOADING CONDITIONS.

Figure 12. Motor-Pump Performance

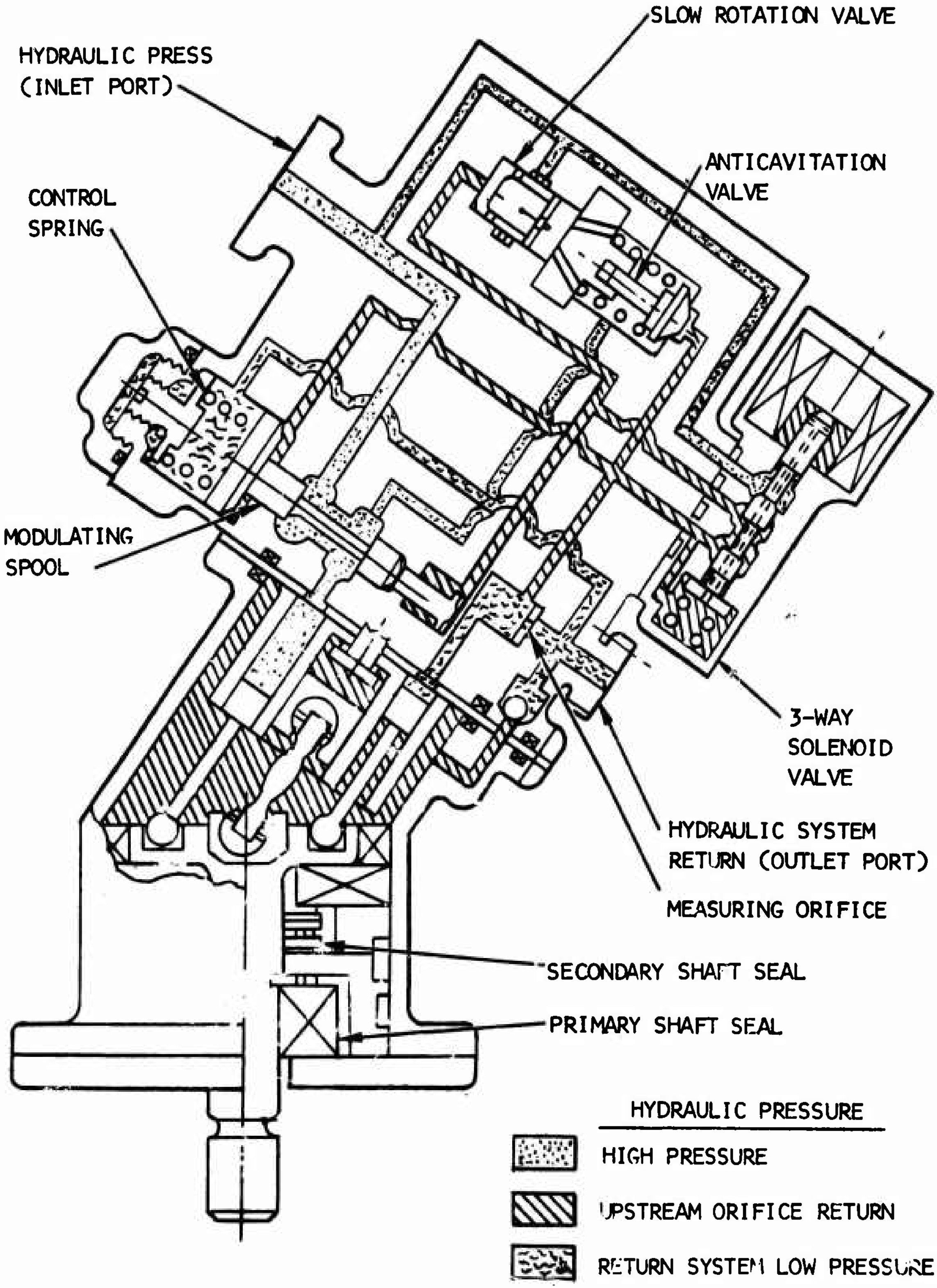


Figure 13. Hydraulic Motor Schematic Diagram

for an actual air vehicle application. It is interesting to note that assuming 60 pounds is a realistic figure for actual air vehicle hardware, this unit's weight is approximately midway between the weights assigned to it for the elevon and main landing gear applications in the Phase I study (54 and 77 pounds, respectively). However, the elevon study of Phase I assumed a 25 percent speed reduction whereas it will be shown later that the actuat hardware only experienced an 8 percent speed reduction. This means that the actual air vehicle unit could be 7 pounds lighter (53 pounds total) and still meet the performance of the study unit. Based on these facts it is quite apparent that actual air vehicly hardware will be lighter in weight than the weight values used for the study.

Section III

TILT TABLE SIMULATOR

A. SIMULATOR

The Tilt Table Simulator is a table which is designed to tilt in a controllable manner, on which a flywheel energy storage substation is mounted. Figure 14 shows the simulator in a partially tilted attitude. Figure 15 is a schematic diagram of the test setup It was used for the investigation and demonstration of flywheel precessional effects on bearings, and to assess the potential gyroscopic effects as related to aircraft stability. An additional principal benefit derived from operation of the tilt table was the experience with the operation of a system of bearings and its associated lubrication system at high speeds.

The tilt table itself is of channel iron construction. The movable platform, on which the flywheel energy storage substation is mounted, is a 16-inch square metal plate 1/2 inch thick. It is pivoted on flange-type ball bearings wherein a 1-inch diameter hole is provided at the bearing centerline axis through which hydraulic lines are routed.

The substation used consists of a motor-pump unit with modified manual control, a gearbox, and a flywheel. The modification of the control permitted the substation to be electrically operated to assume a full motoring mode and a full pumping mode according to the position of the mode and dump valves which are external to the tilt table, but are part of the test setup adjacent to the laboratory hydraulic supply.

A manual pump discharge load valve (see figure 15) downstream of the electrically-operated shutoff valve allowed pressure drop to be regulated at the flow desired to exercise the system at its design capacity. A flow regulator, supplied as part of the motor-pump, is installed in the high-pressure line coming from the laboratory supply to control maximum flywheel speed.

B. PRECESSIONAL LOAD TEST SPECTRUM

A basic objective of the Tilt Table Simulator endurance tests was to show the effects of 3,000 hours of XB-70 airplane flight time on the flywheel bearings in 100 hours of total test time. The analysis of the problem and the spectrum which was established also included other varieties of air vehicles. It is felt that the selection quite well covers the range of performance for manned high-performance air vehicles. Their inclusion and the manner in which they were included provide a scale of values against which the energy storage substation was tested.

The calculated value for bearing life shown in table III and entitled "Percent of Bearing Life Consumed in 3,000 Flight Hours" can be seen to be very low. This could lead to the conclusion that bearings do not represent a problem in flywheel design and would be a correct assumption if the mode of failure were always ball race fatigue.

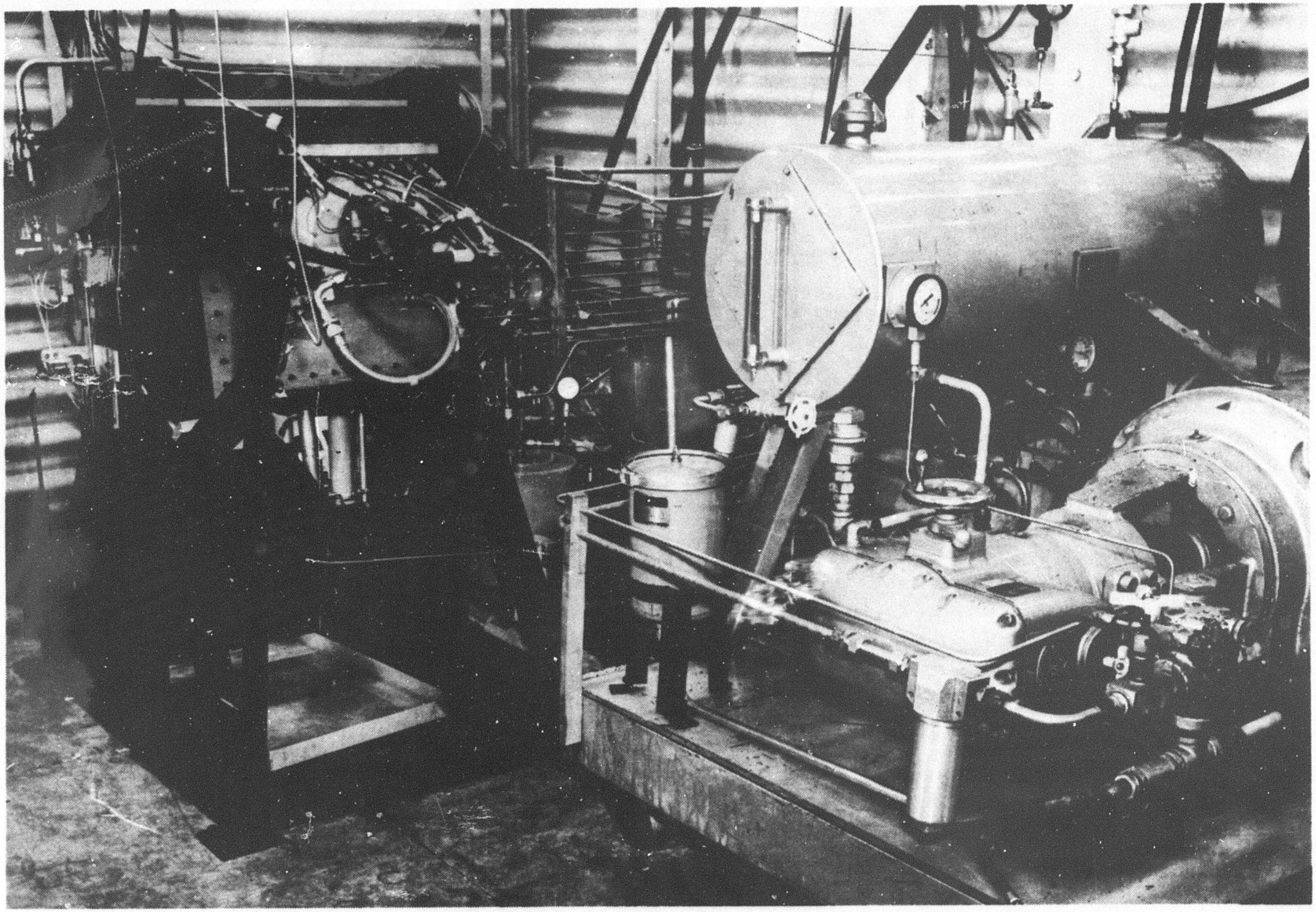
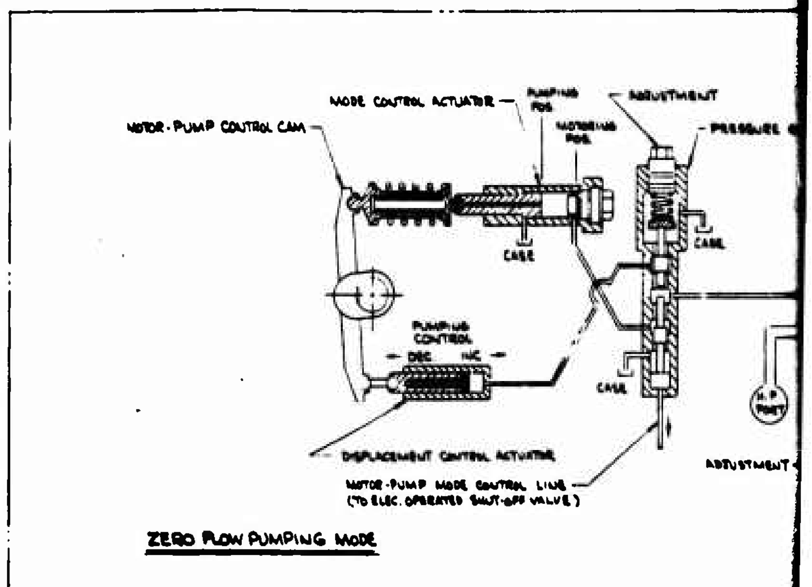
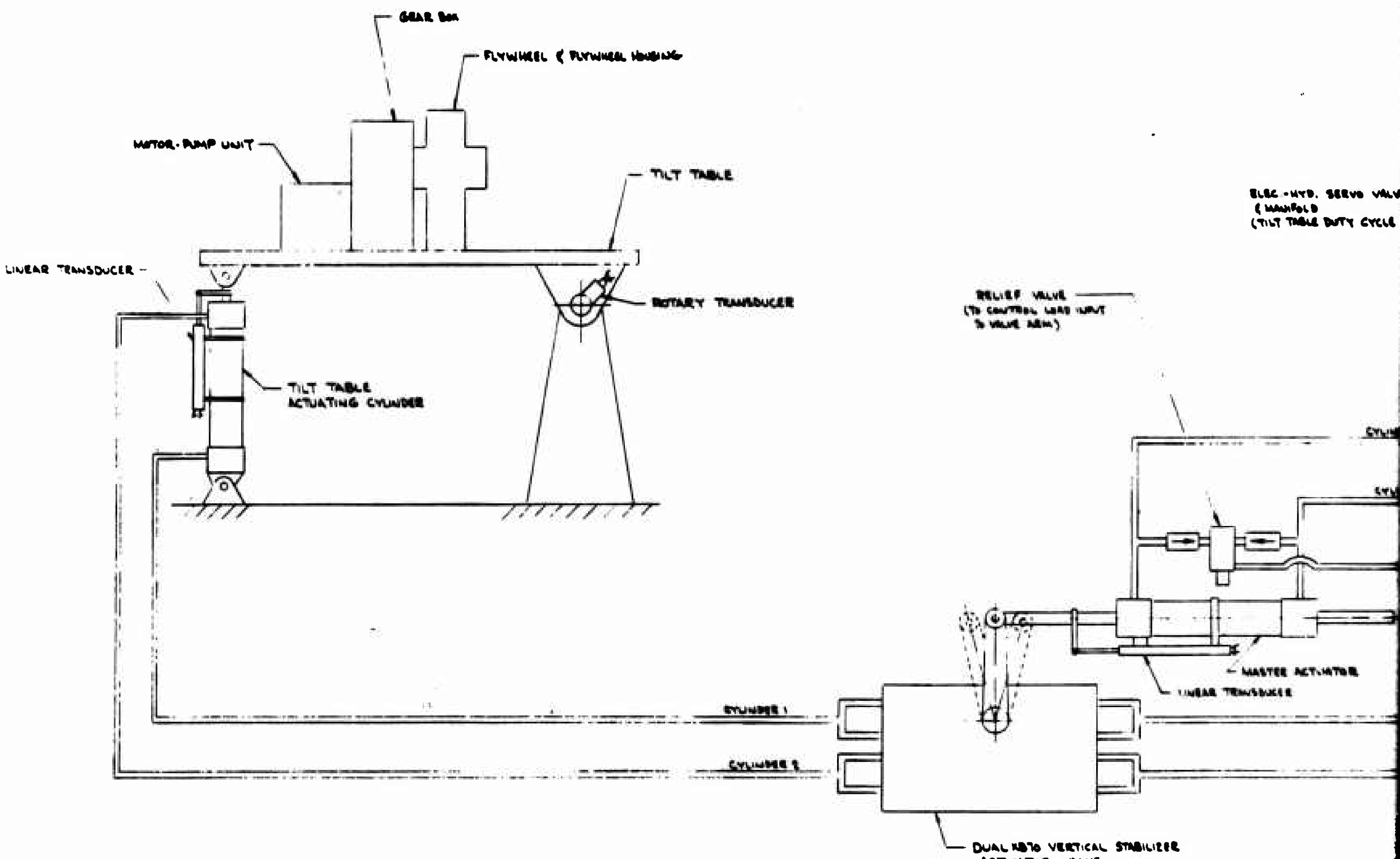


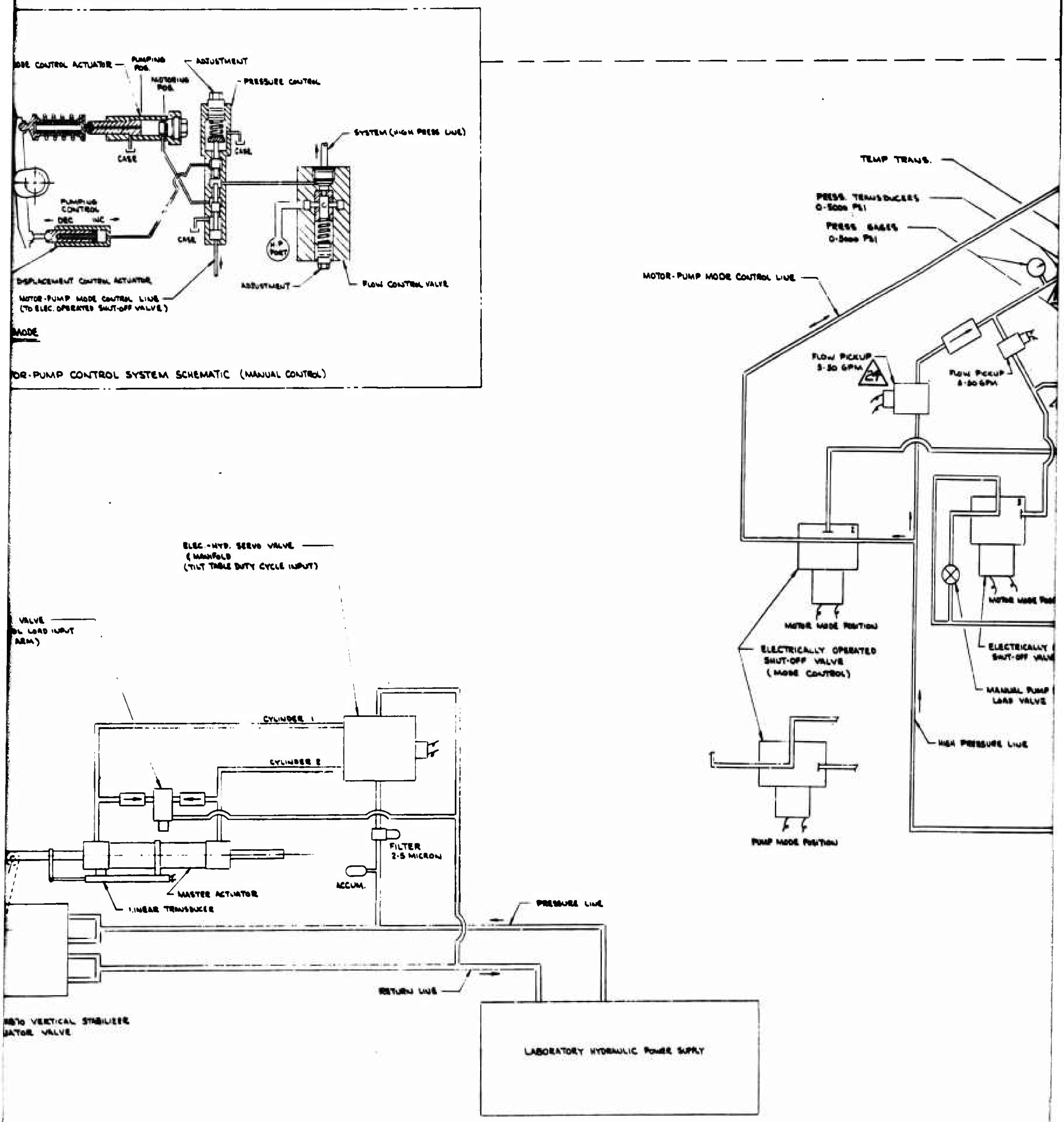
Figure 14. Tilt Table Simulator



MOTOR-PUMP CONTROL SYSTEM SCHEMATIC (MANUAL C

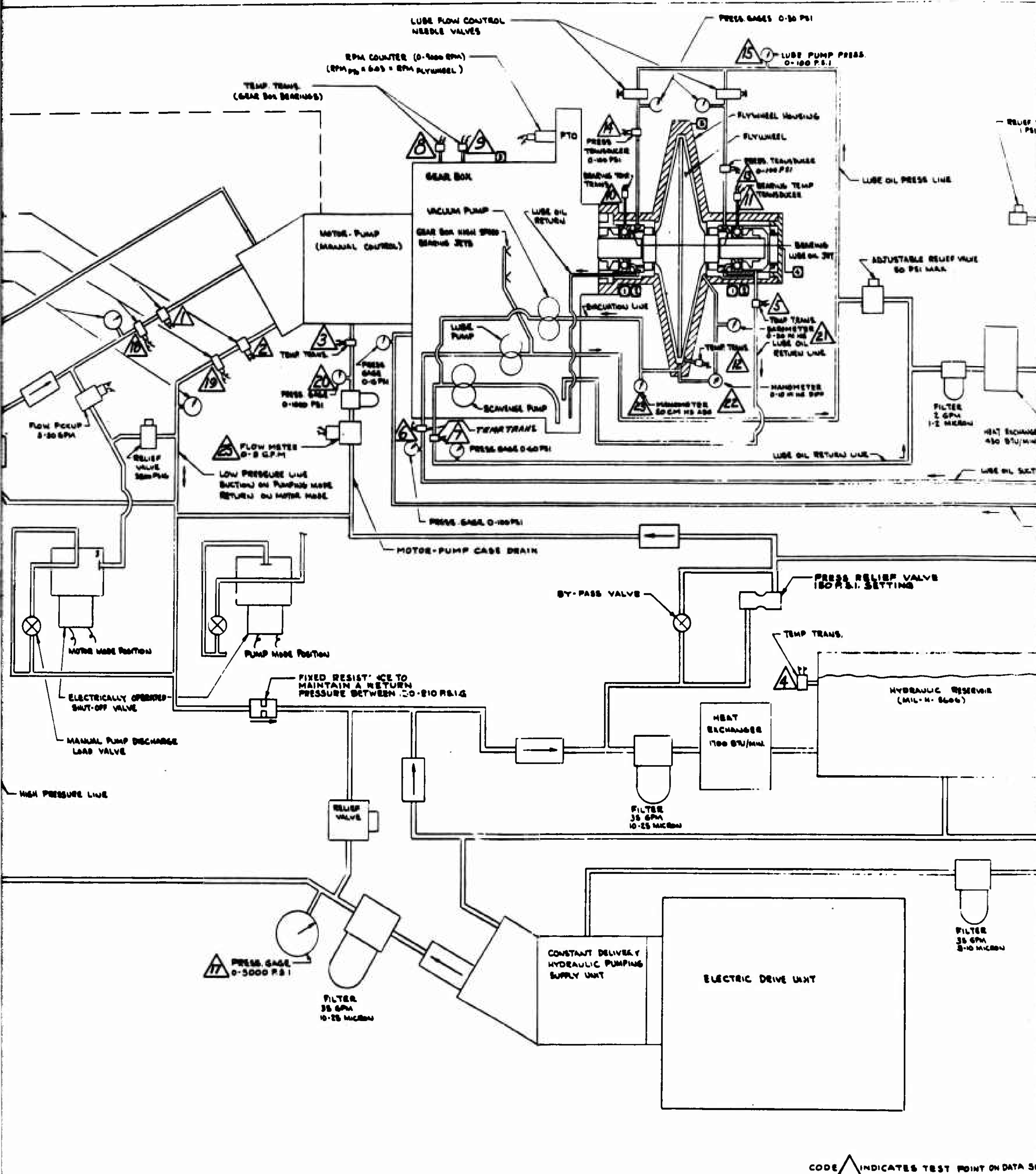


A

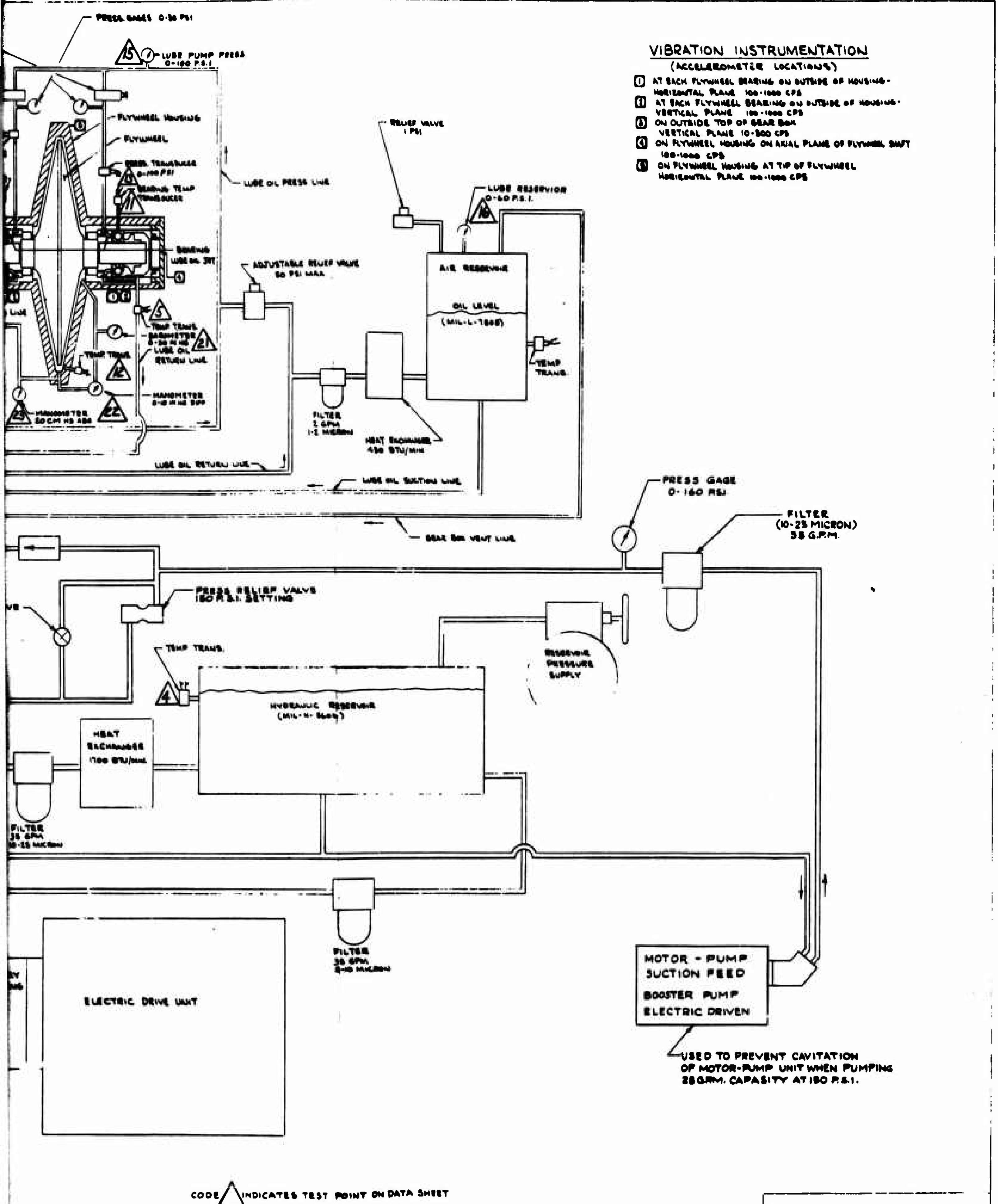


OR-PUMP CONTROL SYSTEM SCHEMATIC (MANUAL CONTROL)

B


LUBE FLOW CONTROL NEEDLE VALVES
PRESS. GAGES 0-30 PSI
LUBE PUMP PRESS. 0-100 P.S.I
RPM COUNTER (0-9000 RPM)
TEMP. TRANS. (GEAR BOX BEARINGS)
PTO
GEAR BOX
FLYWHEEL HOUSING
FLYWHEEL
PRESS. TRANSDUCER 0-100 PSI
BEARING TEMP TRANSDUCER
LUBE OIL PRESS LINE
VACUUM PUMP
LUBE OIL RETURN
GEAR BOX HIGH SPEED BEARING JETS
MOTOR-PUMP (MANUAL CONTROL)
BEARING LUBE OIL JET
ADJUSTABLE RELIEF VALVE 60 PSI MAX.
EVACUATION LINE
LUBE PUMP
TEMP TRANS
LUBE OIL RETURN LINE
MANOMETER
SCAVENGE PUMP
TEMP TRANS
PRESS GAGE 0-60 PSI
FILTER 2 GPM 1-2 MICRON
LUBE OIL RETURN LINE
FLOW PICKUP 3-30 GPM
RELIEF VALVE
FLOW METER 0-3 G.P.M
LOW PRESSURE LINE SUCTION ON PUMPING MODE RETURN ON MOTOR MODE
PRESS GAGE 0-100 PSI
MOTOR-PUMP CASE DRAIN
BY-PASS VALVE
PRESS RELIEF VALVE 150 P.S.I. SETTING
MOTOR MODE POSITION
PUMP MODE POSITION
TEMP TRANS.
ELECTRICALLY OPERATED SHUT-OFF VALVE
FIXED RESISTANCE TO MAINTAIN A RETURN PRESSURE BETWEEN 50-210 P.S.I.G.
HYDRAULIC RESERVOIR
HEAT EXCHANGER 1700 BTU/MIN
MANUAL PUMP DISCHARGE LOAD VALVE
HIGH PRESSURE LINE
RELIEF VALVE
FILTER 35 GPM 10-25 MICRON
FILTER 35 GPM 5-10 MICRON
PRESS. GAGE 0-5000 P.S.I
CONSTANT DELIVERY HYDRAULIC PUMPING SUPPLY UNIT
ELECTRIC DRIVE UNIT
FILTER 35 GPM 10-25 MICRON
CODE △ INDICATES TEST POINT ON DATA SH

C



D

Figure 15. Schematic - Energy Storage Substation Tilt Table

Table III

TILT TABLE TEST TIME AND PRECESSIONAL RATE REQUIREMENTS

Aircraft Group	XB-70		Noted*		Fighter-Reconnaissance	
Maneuver	Pitch	Roll	Pitch	Roll	Pitch	Roll
Maximum precessional rate in maneuver	10 deg/sec	30 deg/sec	10 deg/sec	100 deg/sec	20 deg/sec	180 deg/sec
Precessional rate for tilt table to include a "G" factor	11.7 deg/sec	30.9 deg/sec	11.7 deg/sec	101.7 deg/sec	23.5 deg/sec	182 deg/sec
Precessional load resultant from tilt table rate	54 lb	143 lb	54 lb	470 lb	108 lb	838 lb
Percent of bearing life consumed in 3,000 flight hours	.0265 percent	.0323 percent	.0433 percent	.617 percent	.0577 percent	3.861 percent
Test time required**	38 hr	2.5 hr	62 hr	1.3 hr	10.3 hr	1.48 hr

* Medium size low-level strategic aircraft

** Test time is established on basis that entire test is performed at the precessional rate of line 4.

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Ball race fatigue is the limiting mode of failure for a properly designed, installed, and lubricated bearing which is operated within the load and speed limits for which it was designed. This limiting (longest possible) life can be consistently attained with low speed bearings. However, as bearing speed (DN value) goes up failures, due to other failure modes which are less predictable and controllable, tend to increase in frequency. These other failure modes include, but are not limited to, wear-out failure (usually due to contaminated lubricant), overheat failure (frequently due to improper lubrication system design), and ball separator failure (usually due to improper bearing geometry for the loads and speeds encountered). When these types of failure occur they can, and frequently do, reduce bearing life by several orders of magnitude compared to the bearing life (L) predicted from the above formula.

For the purpose of this program, the problem was attacked by first tabulating pitch, roll, and yaw maneuvers for groups of airplanes which are categorized by their general similarity of flight characteristics (tables IV and V). It will be noted that a table for yaw maneuvers is not included. This is because yaw rates were found to be so small that they could safely be neglected. The airplane groupings are long-range, high-altitude aircraft (XB-70); medium-size, low-level strategic aircraft; and fighter-reconnaissance aircraft. The acceleration and angular velocity characteristics with respect to time for each maneuver was then approximated by a trignometric function, wherein the general characteristic of a sine wave is predominant. An integration of the time at precession rate for each kind of maneuver of a general airplane class was performed and this allowed the construction of the "time exceedance" curves shown in figures 16 through 21.

The time exceedance curves were stepped off in appropriate increments to establish the total percentage of available bearing life that its particular maneuver would use up in the particular aircraft group for which it represents a 3,000-hour flight life. This percentage, together with test time requirements, is listed in table III. The bearing life from which the percentage is proportioned was established by the conventionally accepted equation:

$$L = (C/P)^3$$

wherein:

L = life of bearing

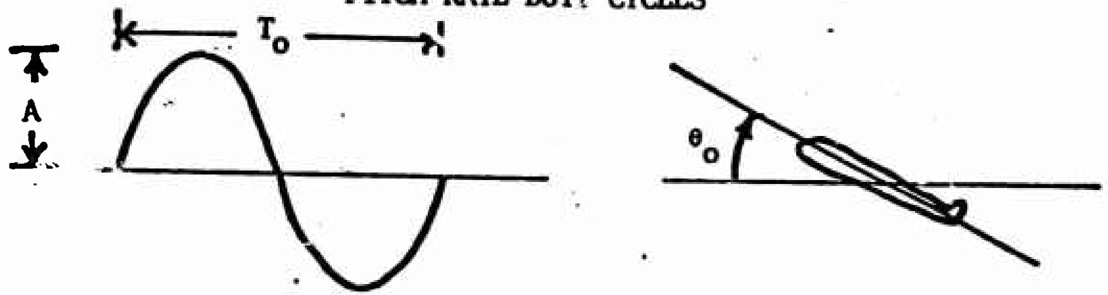
C = a constant established from a known life of the bearing at a known rated load and its design speed

P = load applied to bearing which will result in a life equal to L

It should be noted that the test time requirements could have been listed for several magnitudes of load (i.e., tilt table precessional rate). The one listed represents the maximum precession load plus a G-load increment. In any event, it can readily be seen by inspection that as the load application due to tilt table precession becomes higher, the test time requirement becomes shorter.

Table IV

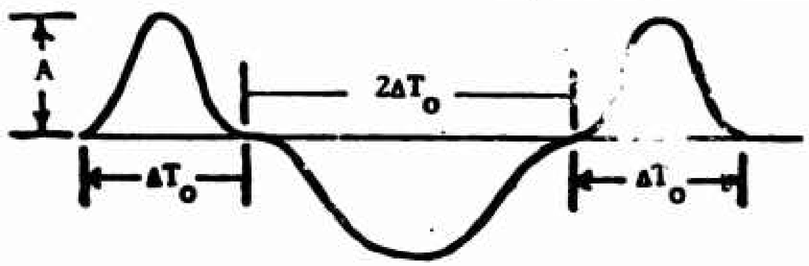
PITCH RATE DUTY CYCLES

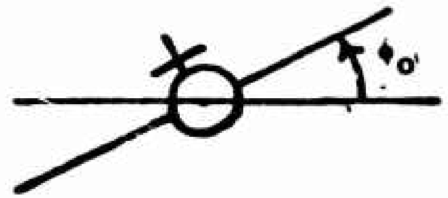


B-70 Missions				Manned Strategic Aircraft Missions				F-R Missions			
A (deg/sec)	T_o (sec)	θ_o (deg)	Reps Per Hour	A (deg/sec)	T_o (sec)	θ_o (deg)	Reps Per Hour	A (deg/sec)	T_o (sec)	θ_o (deg)	Reps Per Hour
10	4	12.7	1	10	4	12.7	1	20	4	25.4	1
8	1	2.6	150	8	1	2.6	250	15	4	19.0	1
5	1	1.6	100	5	1	1.6	200	10	4	12.7	1
5	4	6.4	25	5	4	6.4	25	8	1	2.6	250
[illegible]	4	1.3	25	1	4	1.3	25	5	1	1.6	200
1	8	2.6	12	1	8	2.6	12	5	4	6.4	25
0.5	8	1.3	12	0.5	8	1.3	12	1	4	1.3	25
								1	8	[illegible]	12
								0.5	8	1.3	12

Table V

ROLL RATE DUTY CYCLES





B-70 Missions				Manned Strategic Aircraft Missions				F-R Missions			
A (deg/sec)	T_o (sec)	θ_o (deg)	Reps Per Hour	A (deg/sec)	T_o (sec)	θ_o (deg)	Reps Per Hour	A (deg/sec)	T_o (sec)	θ_o (deg)	Reps Per Hour
30	2	30	0.4	100	1	50	0.6	180	1	90	1.0
15	4	30	2.0	50	2	50	2.0	100	1	50	1.0
10	4	20	2.0	25	4	50	2.0	50	2	50	2.0
5	4	10	2.0	10	4	20	2.0	25	4	50	2.0
2	10	10	2.0	5	4	10	2.0	10	4	20	2.0
1	10	5	5.0	2	10	10	2.0	5	4	10	2.0
0.5	10	2.5	5.0	1	10	5	5.0	2	10	10	2.0
				0.5	10	2.5	5.0	1	10	5	5.0
								0.5	10	2.5	5.0

Dutch roll 1 rep/hour

Dutch roll 2 rep/hr

Dutch roll 2 rep/hr
360-deg rolls 0.4 rep/hr

DUTY CYCLES OF VARIOUS AIRPLANE MODELS FOR FLYWHEEL ANALYSIS
TIME EXCEEDANCE CURVES VERSUS CYCLE AMPLITUDE
SPECTRUM FOR 3,000 FLIGHT HOURS
XB-70 PITCHING CYCLES

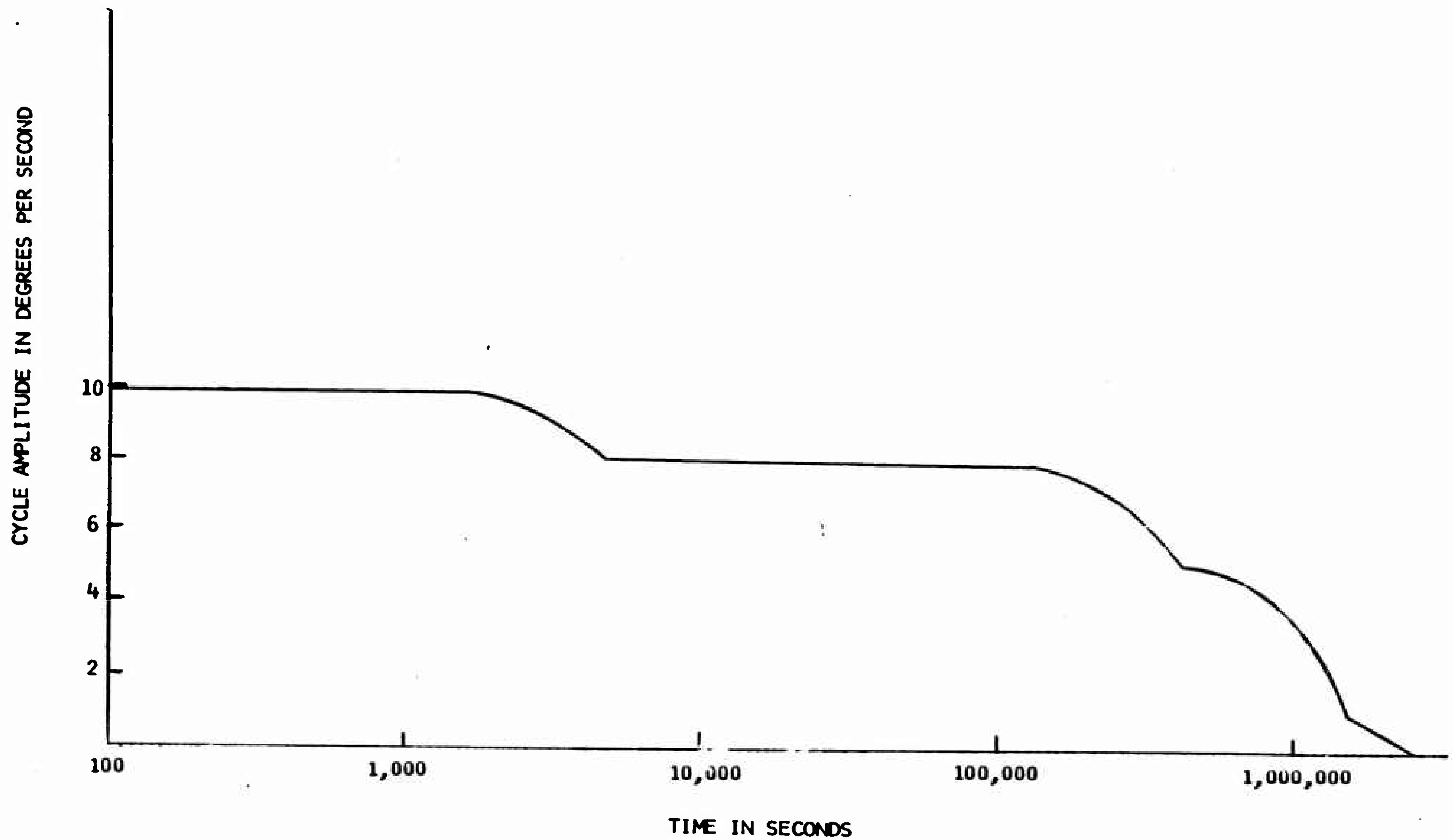


Figure 16. Time Exceedance Curve

DUTY CYCLES OF VARIOUS AIRPLANE MODELS FOR FLYWHEEL ANALYSIS
TIME EXCEEDANCE CURVES VERSUS CYCLE AMPLITUDE
SPECTRUM FOR 3,000 FLIGHT HOURS
XB-70 ROLLING CYCLES

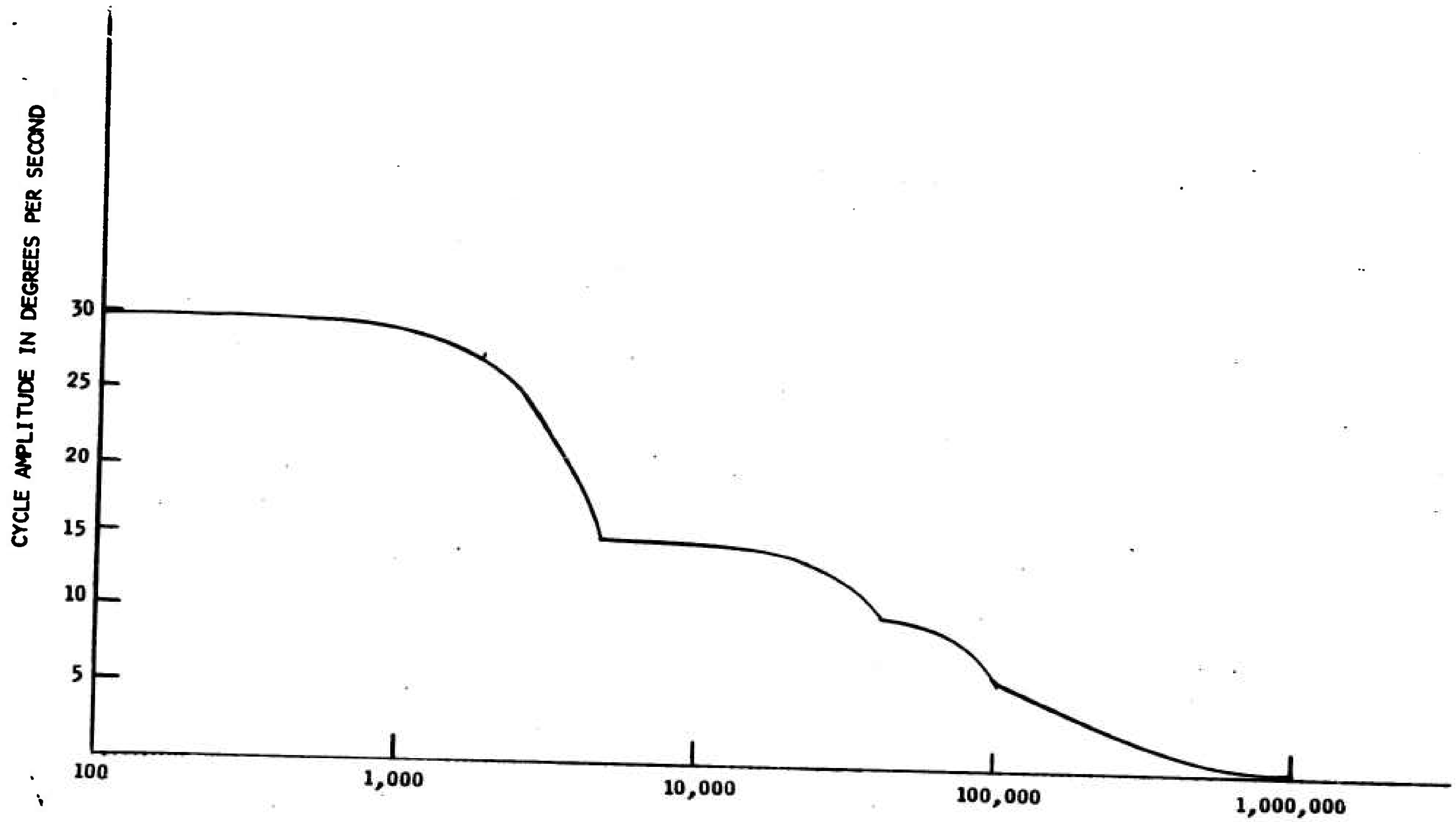


Figure 17. Time Exceedance Curve

DUTY CYCLES OF VARIOUS AIRPLANE MODELS FOR FLYWHEEL ANALYSIS
TIME EXCEEDANCE CURVES VERSUS CYCLE AMPLITUDE
SPECTRUM FOR 3,000 HOURS FLIGHT TIME
MEDIUM SIZE LOW LEVEL STRATEGIC AIRCRAFT PITCHING CYCLES

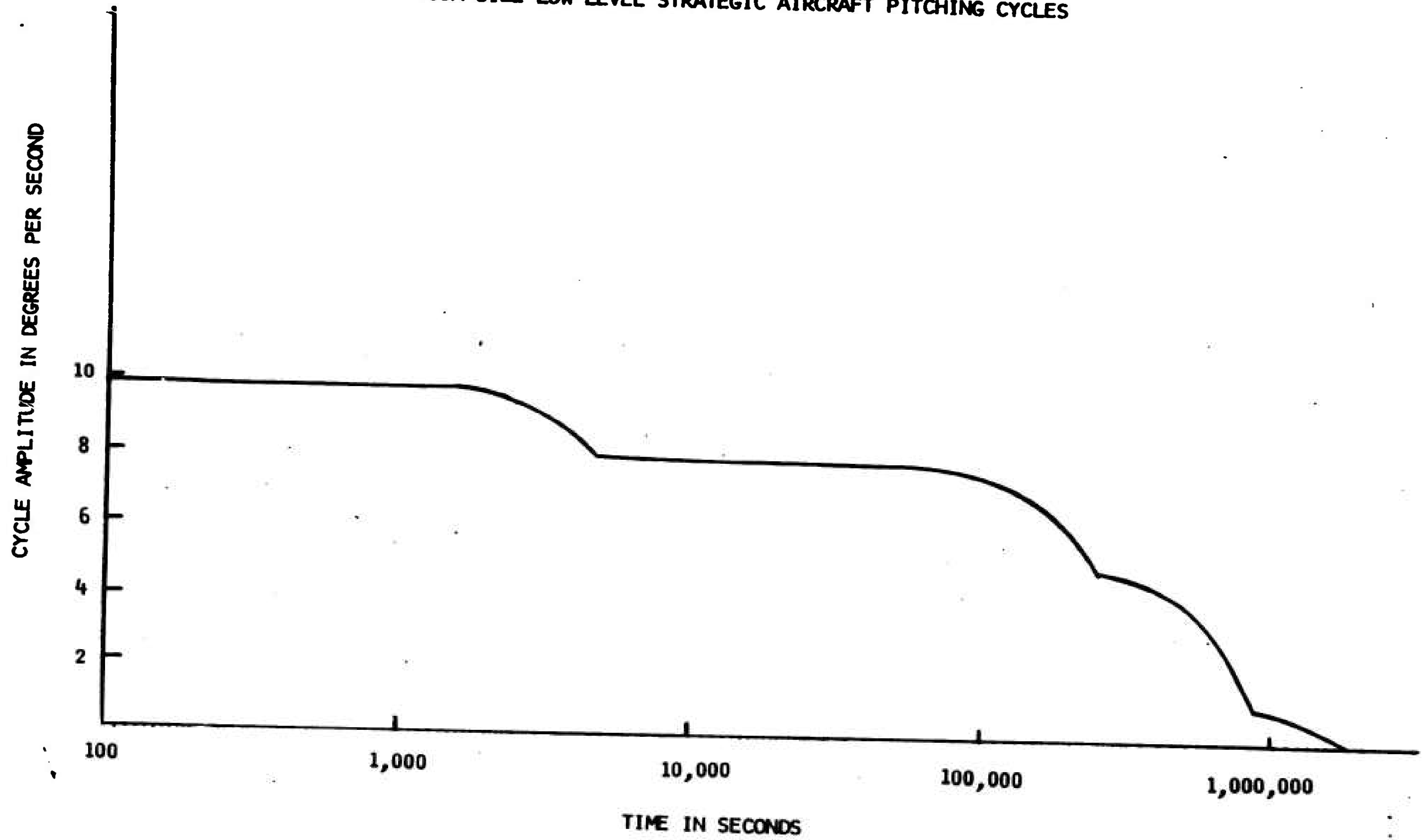


Figure 18. Time Exceedance Curve

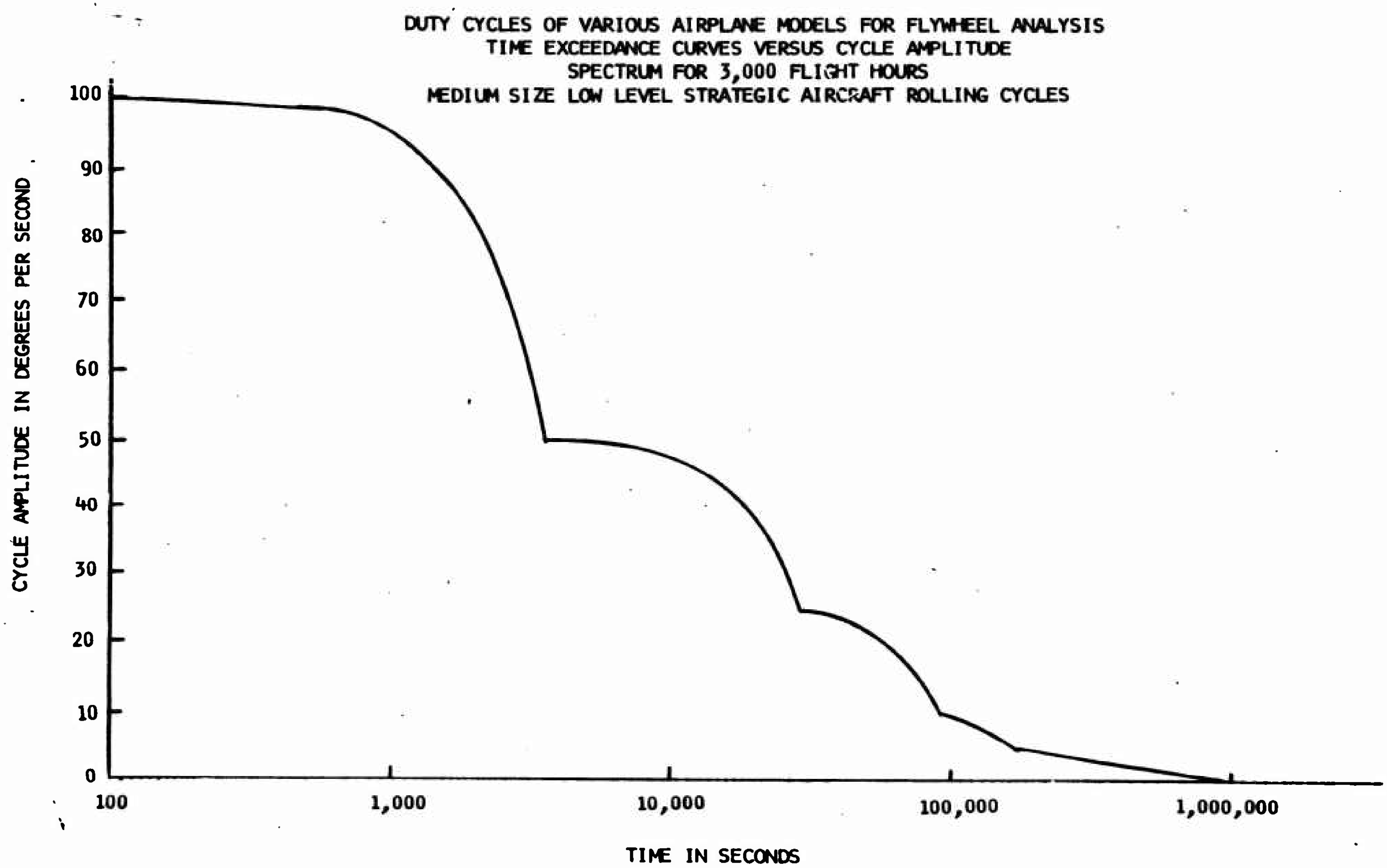


Figure 19. Time Exceedance Curve

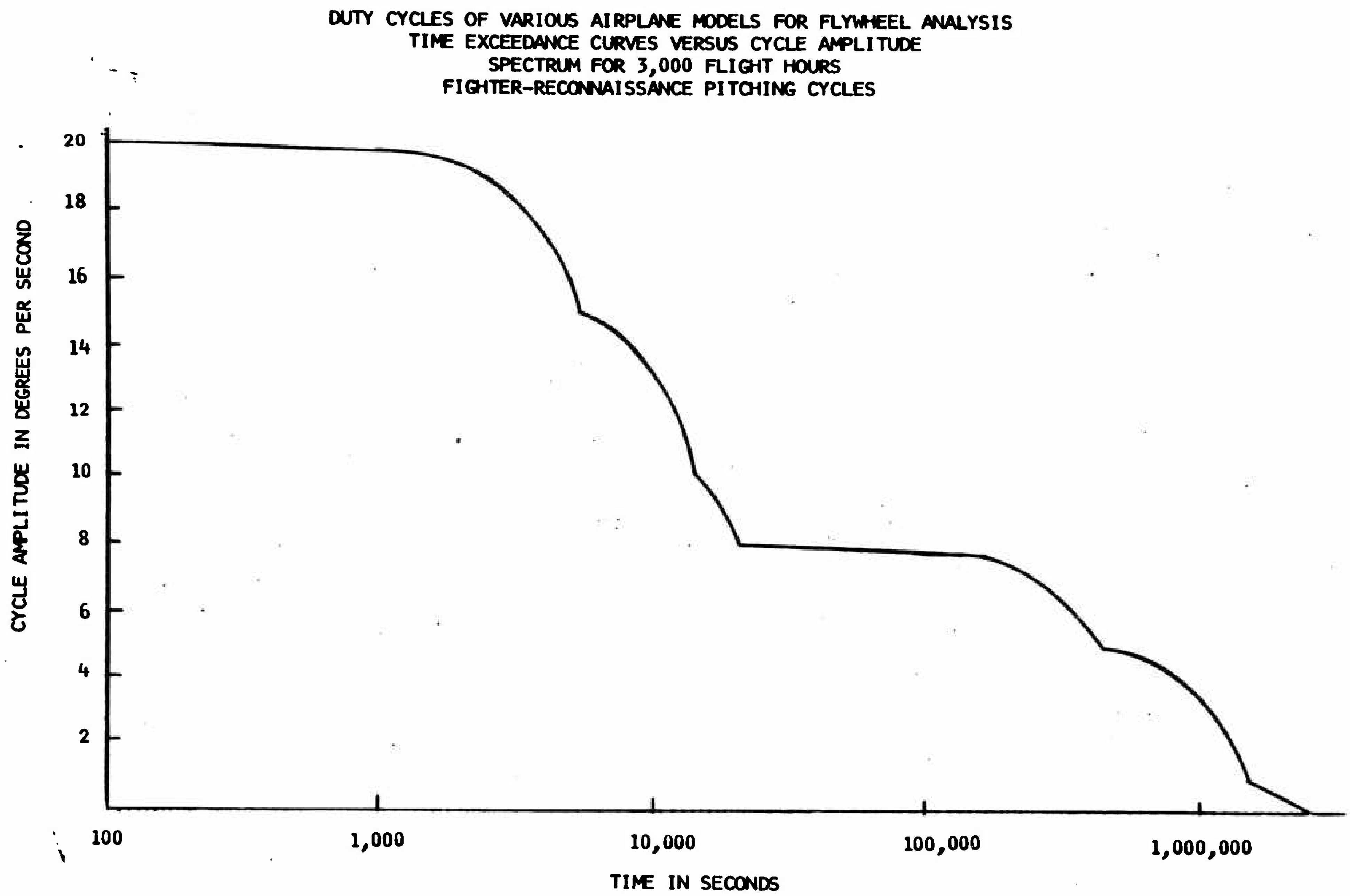


Figure 20. Time Exceedance Curve

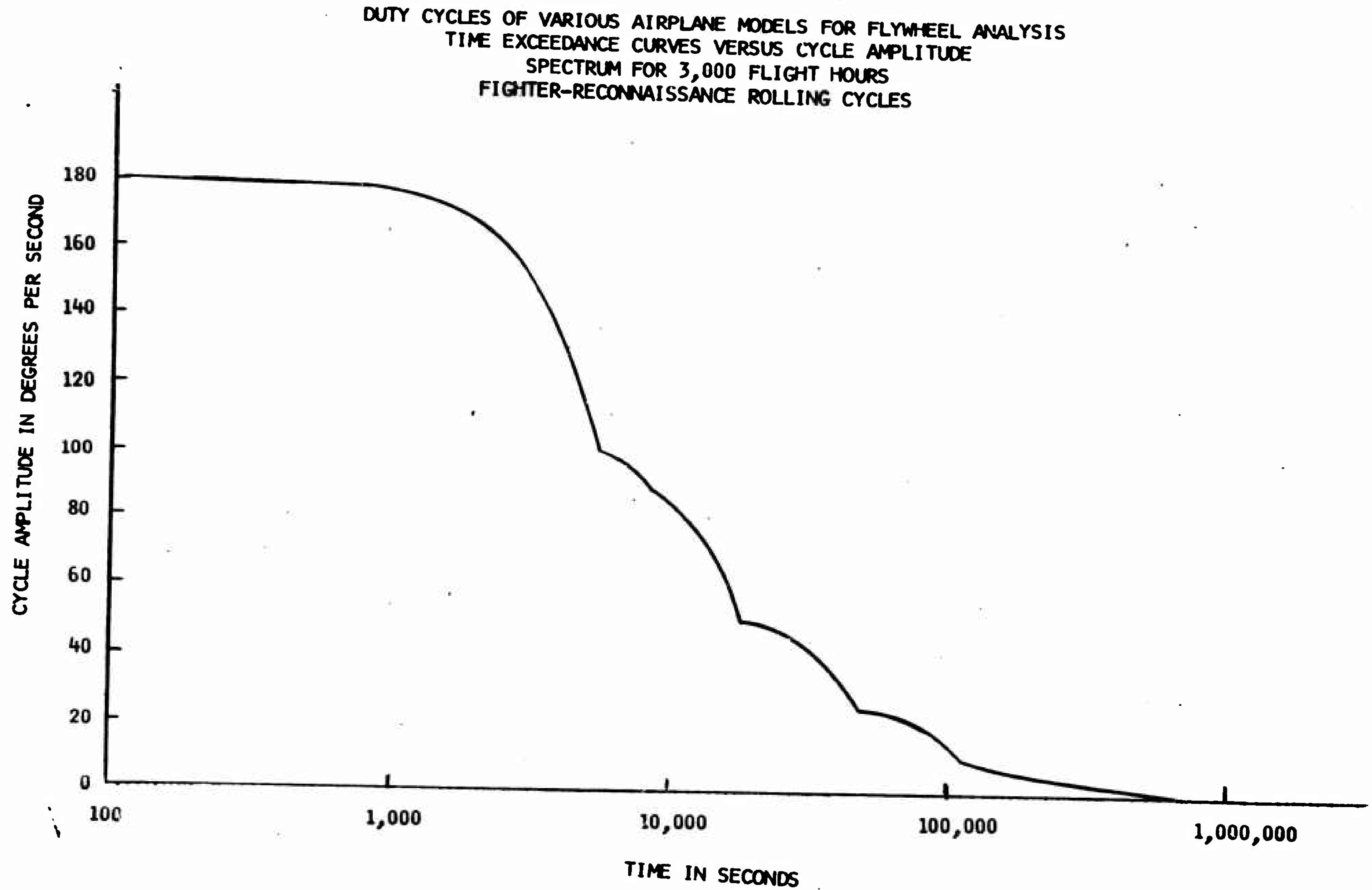


Figure 21. Time Exceedance Curve

As a clarification of the G-load, it should be pointed out that:

1. The G-load never adds algebraically to the precession load except for loads caused by aircraft yaw maneuvers.

2. The precessional loads imposed by yaw maneuvers are negligible. (Since any installation position in an aircraft will always combine them with much larger loads caused by pitch or roll maneuvers, they can be safely neglected.)

3. The G-load varies with the various aircraft maneuvers and aircraft groupings.

4. The G-load adds in a vector quantity for all pitch and roll maneuvers unless the flywheel mounting calls for a vertical axis. In this case, it merely changes the thrust preload on the bearing by adding to one and subtracting from the other.

To satisfactorily account for the above factors, the G-load was added to the precessional load in variable increments arbitrarily chosen, based upon a review of accumulated flight data.

Table VI is basically a listing of the selected test spectrum. It is arranged essentially to allow accomplishment of test objectives in steps which are in ascending order with respect to severity of damage on the bearings, and to give precedence to the fulfillment of XB-70 airplane installation requirements. The test objectives are established considering that the mounting orientation in an airplane will always require that pitch or roll, or both pitch and roll, maneuvers impose precessional loads on the flywheel bearings.

C. TILT TABLE SIMULATOR TESTING

Initial operation of the Tilt Table Simulator substation was begun February 7, 1967. It was planned to increase teststand pressure until the flywheel began to rotate, then gradually increase speed while observing vibration data to determine the frequency, and the magnitude of the resonant points.

When frictional breakout occurred, the speed increased from 0 rpm to over 20,000 rpm and it was found that this was about the minimum speed that could be effectively maintained. This phenomena can be readily explained when it is remembered that it is a characteristic of the design of the New York Airbrake-type motor-pump to display a high breakout friction since the piston shoes do not become floated on a thin film of oil until rotation begins.

Because the speeds between 0 and 20,000 rpm were passed through so quickly, resonant vibration points were observable only as transients. Higher speeds were dwelled upon with sufficient time to establish stabilized conditions where resonant vibration conditions were suspected. The search up to 56,000 rpm was successfully conducted, with a minor delay occurring when 36,000 rpm was reached due to a bearing overheat problem which is described subsequently.

Table VI

TILT TABLE ENDURANCE TEST SPECTRUM

Test Objective		Tilt Table Rate	Bearing Load	Test Time Increment	Time From Start of Test
I	XB-70 pitch	11.7 deg/ sec	54 lb	38 hr	38 hr
II	XB-70 roll	30.9 deg/ sec	143 lb	.45 hr	38.45 hr
III	XB-70 pitch and roll	30.9 deg/ sec	143 lb	2.05 hr	40.5 hr
IV	Noted* and pitch	11.7 deg/ sec	54 lb	0** hr	40.5 hr
V	Fighter-Reconnaissance pitch	23.5 deg/ sec	108 lb	0** hr	40.5 hr
VI	Noted* roll	101.7 deg/ sec	470 lb	1.18 hr	41.68 hr
VII	Noted* pitch and roll	101.7 deg/ sec	470 lb	.09 hr	41.77 hr
VIII	Fighter-Reconnaissance roll	182 deg/ sec	838 lb	1.22 hr	42.99 hr
IX	Fighter-Reconnaissance pitch and roll	182 deg/ sec	838 lb	.02 hr	43.01 hr

* Medium size low-level strategic aircraft.

** This objective was satisfied during test objective III in that the total damage accumulated at the end of III resulting from operating the tilt table for 40.5 hours at 30.9 degrees per second exceeds that from the equivalent of 3,000 hours of operation when the flywheel is mounted so as to be subject to pitching only in the types of air vehicle indicated.

This vibration search revealed no resonance conditions of measurable magnitude that could be positively identified by audible changes in pitch or visual observation.

Figure 22 is an expanded trace of substation vibrational accelerations. This particular trace is one taken from accelerometer No. 5, which was monitored at 35,373 rpm. This trace is typical of the highest acceleration encountered during testing while the substation was operating normally. It can be seen that the trace represents a complex combination of accelerations occurring at assorted high frequencies, the sum of which totals approximately 16.6G.

Because this trace represented one of the worst conditions encountered, it was selected for further analysis. Figure 23 is the result of that analysis. This figure was created by running the magnetic tape results from figure 22 through an analyzer having a five-cycle filter bandwidth. The analyzer scanned frequencies from 0 to 3,000 cycles per second. In effect, the analyzer broke the complex trace down into the equivalent sinusoidal vibrations in terms of frequency and "G" amplitude. From this figure it can be seen that the primary contributor to substation vibration is the motor-pump while the flywheel's contribution is nearly negligible.

Although it was originally felt that vibration pickups would provide an excellent indication of imminent bearing failure, actual testing has not proven this to be true. The high level of motor-pump vibrational output shown in figure 23 successfully masks the changes in bearing vibration level. Therefore, the best indicator of imminent bearing failure remains the temperature pickup.

As previously mentioned, the test operation during resonance search was uneventful until 36,000 rpm was approached. At this speed, unacceptable bearing temperature characteristics appeared. The nature of the problem displayed a time-dependent element, i.e., after a few minutes at or near this speed the temperature began to rise with all the characteristics of a runaway temperature condition. Repeated runs to this speed with varied lube oil flows to measure the cooling effect of the oil came up with the same results. The only exception to this rule occurred when a run was made with the gearbox inadvertently evacuated to a low absolute pressure by the scavenge pump. In this case the outboard bearing ran cool. A comparison of the outboard bearing with the inboard bearing (which always ran cool) indicated the condition could be corrected by adding a bearing cavity vent line to supplement the bearing drain line. This conclusion resulted from the consideration of several factors as follows:

1. The bearings, lube jets, slingers and the internal geometry of the inboard and outboard bearing cavities are identical except for the fact that the inboard slinger (in effect an axial turbine) can blow the air and oil mist it draws through the bearing directly into the gearbox.

2. The outboard slinger blows directly at the end cap and the only egress for the oil-air mixture is through a rather small drain line.

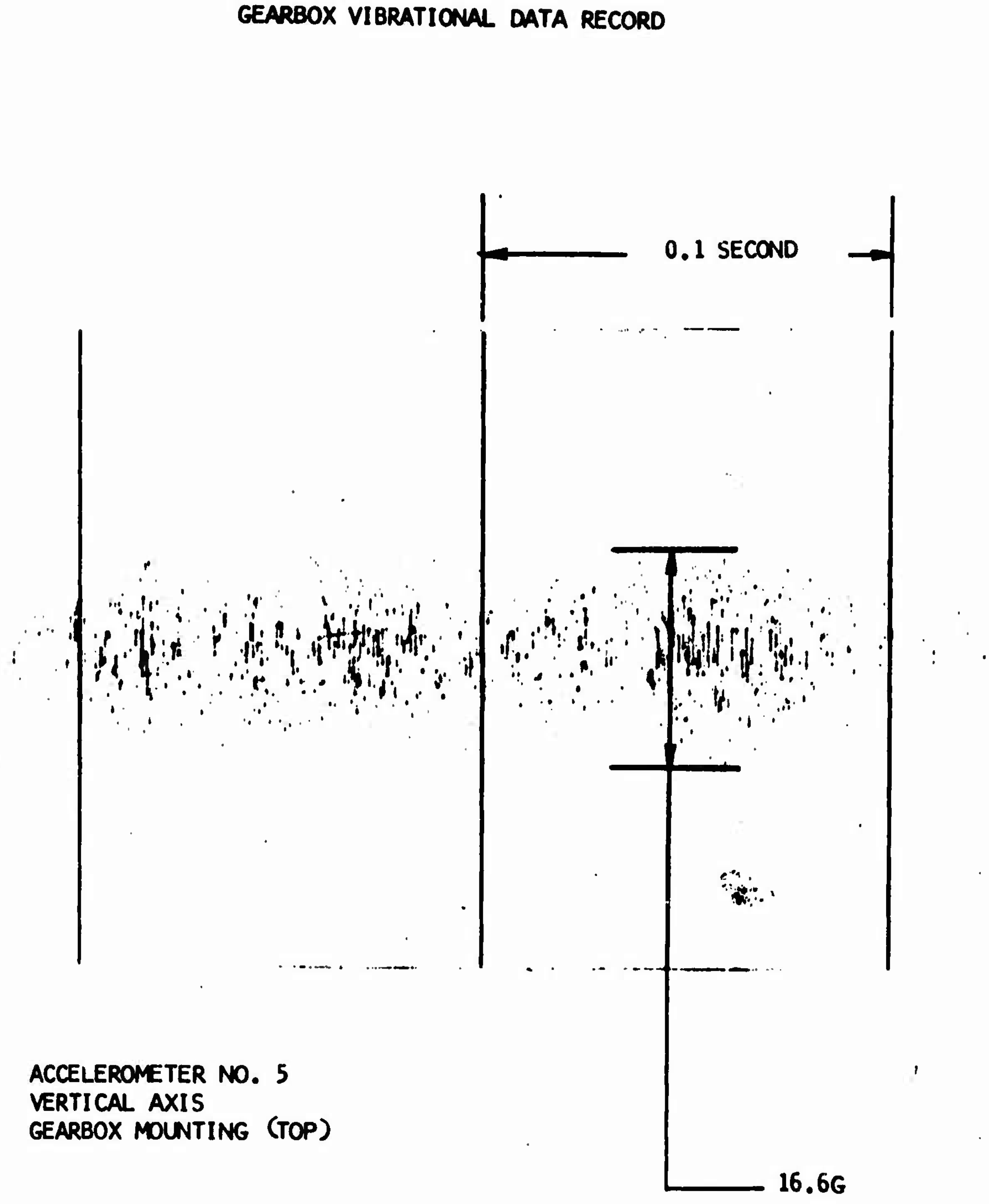


Figure 22. Vibration Trace

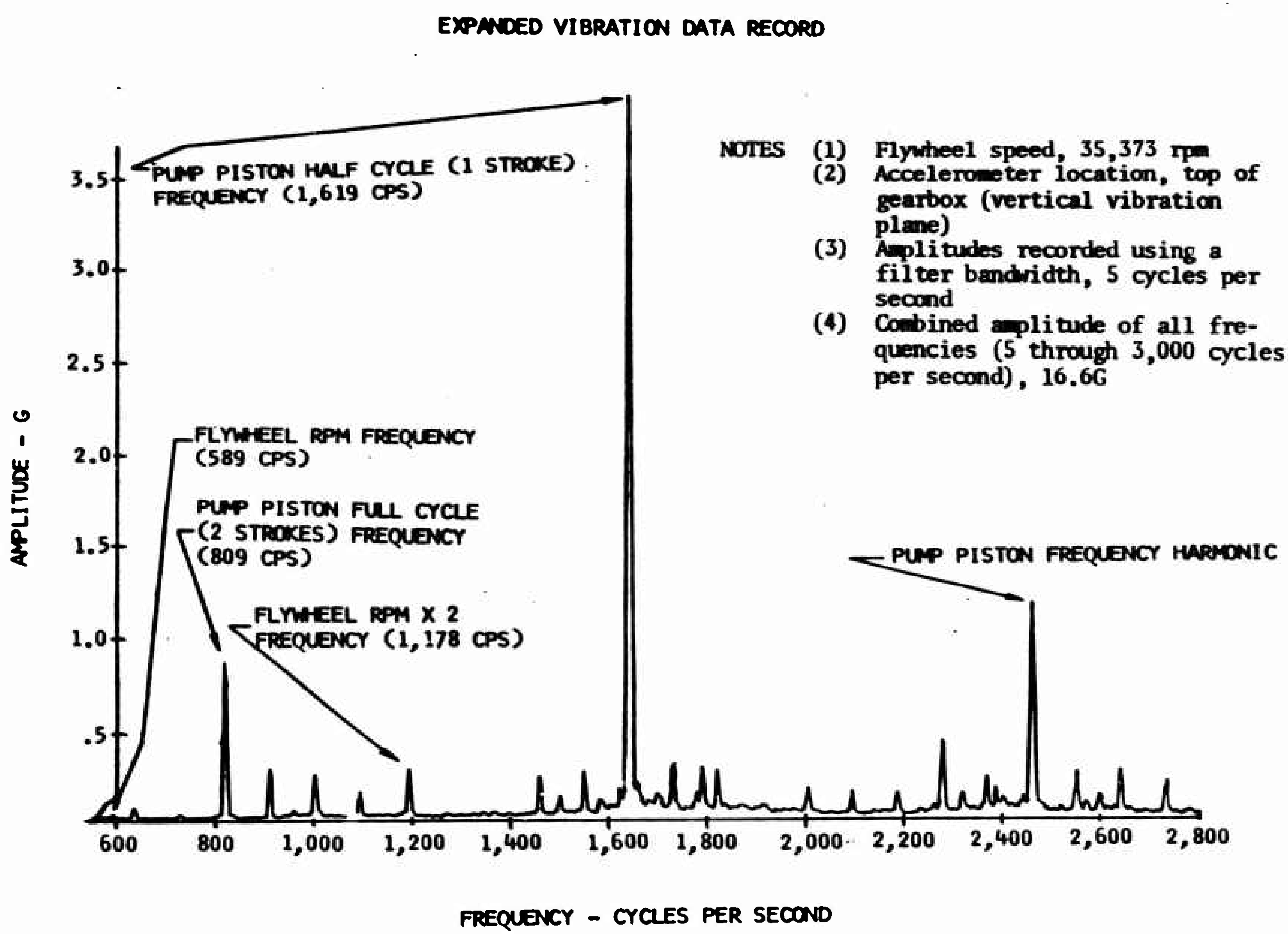


Figure 23. Vibration Analysis

3. Overheat did not occur when the absolute pressure in the bearing cavity was low and the slinger, as a result, was a relatively ineffective air pump.

Based upon the above, it was concluded that the overheating occurred as the result of air recirculation peculiar to the outboard bearing. The air blown by the slinger would find its way back to the low-pressure side of the slinger via the drain passage provided for the carbon face seal coolant oil. This air, already heavily laden with hot oil mist from the bearing, would pick up additional hot oil from the seal and cause the heated and reheated mixture to pass through the bearing a second, third or fourth time. A supplementary effect was to cause the bearing to run flooded, thus generating more heat.

It was correctly concluded that the addition of a vent line would break the recirculating cycle and allow air to enter the cavity tending to help drop the lubrication level. Subsequent operation produced outboard bearing temperatures that were consistent with those of all other gearbox and flywheel bearing temperatures (i.e., 190° to 210° F).

It was noted that when operation was at 52,000 rpm, the flow regulator allowed the motor-pump unit to hunt in 20-second intervals. The flywheel speed varied during this interval from 52,000 rpm to 56,000 rpm. Unsuccessful attempts were made to eliminate the hunting by changing the supply and return pressures.

The motor-pump unit was then cycled to pumping. The resulting slowdown was relatively smooth and quiet. However, toward the end of the cycle, the test stand relief valve started to chatter violently.

A day was spent troubleshooting the relief valve and it was thought that the problem was fixed. However, on the next slowdown cycle the noise and chattering was even more pronounced. Finally, on the third attempted slowdown, the motor-pump destroyed itself. A photograph of the destroyed pump disassembled is shown in figure 24.

At first it was thought that our method of operating the motor-pump during the gradual speed increase incident to resonance search may have initiated the failure. This type of operation involved low inlet pressure (1,500 psi). However, after discussing the failure with New York Airbrake, it was concluded that the failure initiated from a defect in the motor-pump itself. More specifically, it was felt that the failure initiated from gradually-increased binding of a piston which eventually broke that piston's ball end and then complete destruction followed.

Testing was resumed using the XB-70 emergency generator motor, taking advantage of its recirculation loop to perform motor-pump functions with respect to extracting energy from the flywheel. The power losses associated with the recirculation loop consume a modest amount of power. The power consumed was sufficient so that a 15 percent flywheel slowdown occurred in less than 20 seconds. However, it was not considered feasible to further modify the control head so that greater amounts of power could be consumed.

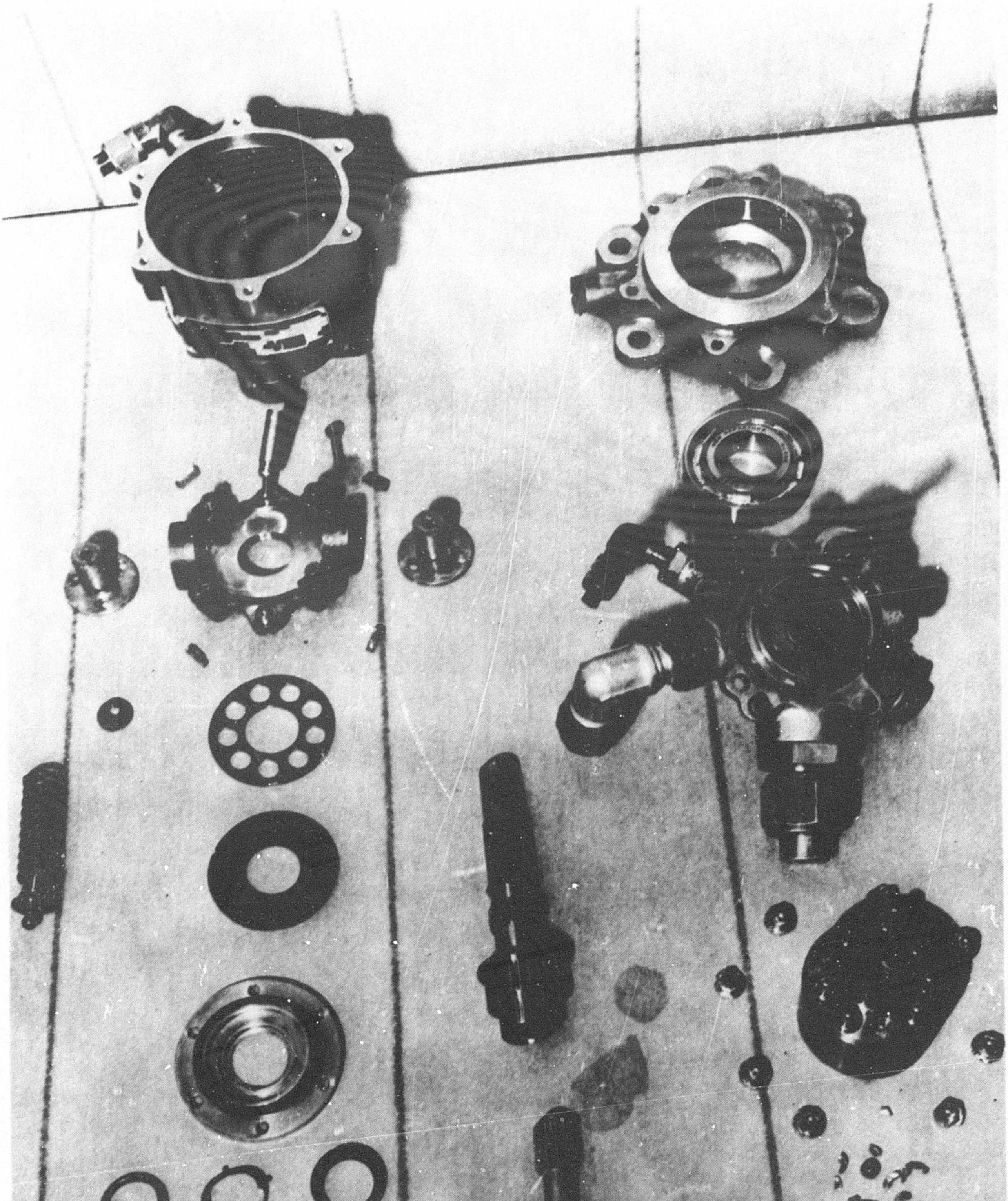


Figure 24. Destroyed Motor-Pump

Recognizing that it is a basic purpose of the tilt table tests to torque load the flywheel while subjecting it to precessional loads, it follows that the reduced slowdown rate and its associated fatigue damage could be compensated for by using increased power for acceleration. To exactly duplicate the torque loading characteristics of the original motor-pump, assuming essentially equal acceleration and deceleration torques, would require a supply pressure of 1,850 psi for the substitute motor-pump. This was determined from the following relationship:

$$3000 \text{ psi} \times \frac{.4 \text{ in.}^3\text{/rev(Original Motor-Pump Unit)}}{.646 \text{ in.}^3\text{/rev(Substitute Motor Unit)}} = 1850 \text{ psi}$$

However, to compensate for the substitute motor-pump's lack of full deceleration torque capabilities, the acceleration torque was increased by raising the supply pressure to 2,000 psi.

The endurance test was begun with the motor-pump set up as just described. The test followed the spectrum of table VI. Approximately 34.5 hours of testing was completed out of 38 hours required for completion of Test Objective I in the spectrum. Table VIII (nine pages) is the recorded data. Table VIII is a summary of calculated results. Figure 25 is a sample data record from which some of the data were taken, and which was used to monitor the performance of the test.

The first 24 hours of testing was conducted with the tilt rate adjusted to slightly less than 12 degrees per second. The tilt rate had some variation throughout the stroke. Maximum tilt rate had a random relationship to instantaneous flywheel speeds, since this was a function of an independent flywheel speedup and slowdown cycle. The net result of these factors was to create bearing precessional loads of a variety of magnitudes. The accumulated data indicated that the average magnitude represented a reduced load on the bearings when compared to the constant 54 pounds called for in the test spectrum.

At this point (24 hours) in the test, a review of the effect of this reduced load was made. It was determined that an increased load of 61 pounds for the remaining 14 hours in Test Objective I would result in a bearing life usage equivalent to the original intent (i.e., that of 38 hours with a 54-pound bearing load).

The actual application of the newly increased rate resulted in a load that fluctuated between 59 and 64 pounds. This load, although it fluctuated, remained more consistent in the remaining portion of the test because the flywheel speedup, slowdown cycles were discontinued.

Shortly before the end of the last full day's testing (30 hours) the vibration traces began to act peculiarly. The test stand was shut down and it was found that several accelerometer leads were loose. After tightening these leads, the test stand was restarted and the vibration traces were much improved; however, they did not return completely to their original characteristics. At the time, the continued peculiarity of the traces was ascribed to faulty pickups; however, in retrospect, it is now known that this signaled the start of serious bearing deterioration.

Table VII

TILT TABLE SIMULATOR TEST DATA

Summary Data Sheet No. 1 | Date: 2-28-67

NO.									
	Running Time of Test - Hours	0.5	1.0	1.5	2.0	2.5	3.0	3.5	4.0
	Vibration Amp. at Flywheel Bearings-g's — Inboard	4.1	4.1	4.1	4.0	4.1	4.1	4.1	4.1
	Vibration Amp. at Flywheel Bearings-g's — Outboard	3.8	3.8	3.8	3.7	3.9	3.4	3.7	3.7
	(1) Flywheel Speed at Full Motoring-RPM	52,000	52,000	52,000	52,000	52,000	52,000	52,000	52,000
	(2) Minimum Speed Attain Under Load-RPM	44,500	44,500	44,500	44,500	44,500	44,500	44,500	44,500
	Total Precessional Load Cycles	200	410	615	841	1020	1231	1436	1652
	Ave. Tilt Table Velocity - deg/sec.	11.7	12.3	11.8	11.8	11.5	11.8	12.0	11.7
	Ave. Load on Flywheel Bearings - lbs.	41	40	40	42	41	42	42	41
NO.	TEMPERATURE DATA - DEG. F								
1	Test Pump Inlet	130	135	135	147	138	135	135	133
2	Test Pump Return	135	140	140	147	148	152	138	138
3	Test Pump Case Drain	201	205	208	215	200	198	212	208
4	Main Hyd. Reservoir Supply	125	130	130	130	132	130	128	128
5	Lube Oil Return	187	195	198	201	203	203	201	203
6	Lube Oil Suction	80	85	88	90	90	90	90	89
7	Scavenge Oil Return	130	135	132	135	135	138	132	132
8	Gear Box Bearing Adjacent Pump	195	200	200	202	200	200	201	200
9	Gear Box Bearing Adjacent Flywheel	185	185	185	182	185	185	185	183
10	Inboard Flywheel Bearing	195	205	208	210	209	208	208	210
11	Outboard Flywheel Bearing	185	195	195	198	198	198	198	198
12	Flywheel Case Housing	152	170	172	178	175	178	178	176
NO.	SYSTEM PRESSURE DATA - PSIG								
13	Outboard Flywheel Bearing	26	26	26	26	27	27	27	27
14	Inboard Flywheel Bearing	27	27	27	27	28	28	28	28
15	Lube Pump Pressure Port	68	68	68	68	67	65	65	65
16	Lube Reservoir	0	0	0	0	0	0	0	0
17	Hyd. Supply Pump Press. Port	2150	2150	2150	2150	2150	2150	2150	2150
18	Test Motor Press. Port	2000	2000	2000	2000	1980	2000	1990	2000
19	Test Motor Return Port	110	110	110	110	110	110	110	110
20	Test Motor Case Drain	4	4	4	4	4	4	4	4
21	Vacuum at Flywheel Hub - in. Hg	29.5	29.5	29.6	29.5	29.5	29.5	29.4	29.4
22	Vacuum Across Flywheel Housing (Hub to Tip ΔP) - in. Hg	1.5	1.6	1.6	1.6	1.5	1.5	1.5	1.5
23	Press at Flywheel Hub - cm Hg Ab	2.3	2.4	2.4	2.4	2.4	2.4	2.3	2.3
NO.	FLUID FLOW DATA - GPM								
24	Inlet Flow to Motor	23	23	23	23	23	23	23	23
25	Case Drain Flow of Motor	.2to.4	.2-.4	.2-.4	.2-.4	.2-.4	.2-.4	.2-.4	.2-.4
26	Outboard Bearing Lube Flow cc/min.	400	400	400	400	405	405	405	405
27	Inboard Bearing Lube Flow cc/min.	405	405	405	405	410	410	410	410

Table VII (Continued)

TILT TABLE SIMULATOR TEST DATA

Summary Data Sheet No. 2 Date: 2-28-67

	Running Time of Test - Hours		4.5	5.0	5.5	6.0	6.5	7.0	7.5	8.0
	Vibration Amp. at Flywheel Bearings-g's	Inboard	4.4	4.3	4.5	4.3	4.6	4.6	4.6	4.8
		Outboard	4.7	4.7	4.8	4.7	4.9	4.9	4.9	5.2
	(1) Flywheel Speed at Full Motoring-RPM		52,000	52,000	52,000	52,000	52,000	52,000	52,000	52,000
	(2) Minimum Speed Attain Under Load-RPM		44,500	44,500	44,500	44,500	44,500	44,500	44,500	44,500
	Total Precessional Load Cycles		1855	2080	2280	2480	2680	2885	3085	3294
	Ave. Tilt Table Velocity - deg/sec.		11.9	11.6	11.7	11.2	11.6	11.6	11.6	11.6
	Ave. Load on Flywheel Bearings - lbs.		41	42	41	41	43	42	42	42
NO.	TEMPERATURE DATA - DEG. F									
1	Test Pump Inlet		135	135	135	133	132	131	135	132
2	Test Pump Return		138	138	138	137	136	133	136	137
3	Test Pump Case Drain		208	200	205	211	208	210	204	208
4	Main Hyd. Reservoir Supply		128	128	128	128	125	125	125	126
5	Lube Oil Return		201	202	204	200	201	200	201	200
6	Lube Oil Suction		89	89	89	90	88	88	88	88
7	Scavange Oil Return		132	133	134	132	130	132	131	130
8	Gear Box Bearing Adjacent Pump		200	200	200	200	200	200	201	201
9	Gear Box Bearing Adjacent Flywheel		185	185	182	180	180	180	182	182
10	Inboard Flywheel Bearing		210	209	208	204	208	208	207	205
11	Outboard Flywheel Bearing		198	198	200	201	198	198	199	195
12	Flywheel Case Housing		178	174	178	175	172	172	175	172
NO.	SYSTEM PRESSURE DATA - PSIG									
13	Outboard Flywheel Bearing		27	27	27	27	27	27	27	27
14	Inboard Flywheel Bearing		28	28	28	28	28	28	28	28
15	Lube Pump Pressure Port		68	68	68	68	68	68	68	68
16	Lube Reservoir		0	0	0	0	0	0	0	0
17	Hyd. Supply Pump Press. Port		2150	2150	2150	2150	2150	2150	2150	2150
18	Test Motor Press. Port		2000	2000	2000	2000	2000	1990	2000	2000
19	Test Motor Return Port		110	110	110	110	110	110	110	110
20	Test Motor Case Drain		4	4	4	4	4	4	4	4
21	Vacuum at Flywheel Hub - in. Hg		29.4	29.4	29.4	29.4	29.4	29.4	29.4	29.4
22	(Hub to Tip ΔP) Vacuum Across Flywheel Housing - in. Hg		1.5	1.5	1.5	1.5	1.5	1.5	1.5	1.5
23	Press. at Flywheel Hub - cm Hg Ab		2.3	2.3	2.2	2.2	2.2	2.2	2.2	2.2
NO.	FLUID FLOW DATA - GPM									
24	Inlet Flow to Motor		23	23	23	23	23	23	23	23
25	Case Drain Flow of Motor		.2-.4	.2-.4	.2-.4	.2-.4	.2-.4	.2-.4	.2-.4	.2-.4
26	Outboard Bearing Lube Flow	cc/min	405	405	405	405	405	405	405	405
27	Inboard Bearing Lube Flow	cc/min	410	410	410	410	410	410	410	410

Table VII (Continued)

TILT TABLE SIMULATOR TEST DATA

Summary Data Sheet No. 3 Date: 2-28-67

Running Time of Test - Hours	8.5	9.0	9.5	10.0	10.5	11.0	11.5	12.0
Vibration Amp. at Flywheel Bearings-g's — Inboard	4.3	4.9	5.0	4.8	5.0	5.0	4.6	4.9
Vibration Amp. at Flywheel Bearings-g's — Outboard	4.9	5.3	5.0	5.3	5.2	5.2	5.0	5.1
(1) Flywheel Speed at Full Motoring-RPM	52,000	52,000	52,000	52,000	52,000	52,000	52,000	52,000
(2) Minimum Speed Attain Under Load-RPM	44,500	44,500	44,500	44,500	44,500	44,500	44,500	44,500
Total Processional Load Cycles	3509	3722	3941	4154	4380	4600	4830	5050
Ave. Tilt Table Velocity - deg/sec.	12.0	12.0	12.1	12.4	12.1	12.3	12.3	12.4
Ave. Load on Flywheel Bearings - lbs.	41	42	43	44	42	42	43	43
NO. TEMPERATURE DATA - DEG. F								
1 Test Pump Inlet	132	132	133	134	132	132	131	131
2 Test Pump Return	136	133	138	138	133	136	135	136
3 Test Pump Case Drain	210	212	201	210	214	201	208	208
4 Main Hyd. Reservoir Supply	126	128	126	128	128	125	122	124
5 Lube Oil Return	202	205	202	202	205	203	205	203
6 Lube Oil Suction	88	88	89	89	88	88	88	85
7 Scavange Oil Return	129	130	131	132	131	130	130	128
8 Gear Box Bearing Adjacent Pump	202	204	201	204	202	202	204	202
9 Gear Box Bearing Adjacent Flywheel	183	185	185	184	185	183	184	182
10 Inboard Flywheel Bearing	207	208	208	210	209	209	210	208
11 Outboard Flywheel Bearing	191	198	198	198	198	198	198	198
12 Flywheel Case Housing	172	171	171	171	172	171	171	170
NO. SYSTEM PRESSURE DATA - PSIG								
13 Outboard Flywheel Bearing	27	27	27	27	27	28	28	28
14 Inboard Flywheel Bearing	28	28	28	28	28	28	28	28
15 Lube Pump Pressure Port	68	68	68	68	68	68	68	68
16 Lube Reservoir	0	0	0	0	0	0	0	0
17 Hyd. Supply Pump Press. Port	2150	2150	2150	2150	2150	2150	2150	2150
18 Test Motor Press. Port	1990	1990	2000	2000	2000	2000	2000	2000
19 Test Motor Return Port	110	110	110	110	110	110	110	110
20 Test Motor Case Drain	4	4	4	4	4	4	4	4
21 Vacuum at Flywheel Hub - in. Hg	29.4	29.4	29.4	29.4	29.4	29.4	29.4	29.4
22 (Hub to TIP ΔP) Vacuum Across Flywheel Housing - in.Hg	1.5	1.5	1.5	1.4	1.4	1.4	1.4	1.4
23 Press at Flywheel Hub - cm. Hg Ab	2.2	2.2	2.2	2.2	2.3	2.3	2.3	2.3
NO. FLUID FLOW DATA - GPM								
24 Inlet Flow to Motor	23	23	23	23	23	23	23	23
25 Case Drain Flow of Motor	.2-.4	.2-.4	.2-.4	.2-.4	.2-.4	.2-.4	.2-.4	.2-.4
26 Outboard Bearing Lube Flow cc/min	405	405	405	405	405	410	410	410
27 Inboard Bearing Lube Flow cc/min	410	410	410	410	410	410	410	410

Table VII (Continued)

TILT TABLE SIMULATOR TEST DATA

Summary Data Sheet No. 4 | Date: 5-[illegible]-67

NO.	Item								
	Running Time of Test - Hours	12.5	13.0	*13.5	14.0	14.5	15.0	15.5	16.0
	Vibration Amp. at Flywheel Bearings-g's — Inboard	4.8	4.9	-	-	-	-	-	-
	Vibration Amp. at Flywheel Bearings-g's — Outboard	5.3	5.3	-	-	-	-	-	-
	(1) Flywheel Speed at Full Motoring-RPM	52,000	52,000	52,000	52,000	52,000	52,000	52,000	52,000
	(2) Minimum Speed Attain Under Load-RPM	44,500	44,500	44,500	44,500	44,500	44,500	44,500	44,500
	Total Precessional Load Cycles	5262	5487	5685	5886	6105	6322	6530	6744
	Ave. Tilt Table Velocity - deg/sec.	12.0	11.7	11.9	11.9	12.0	11.9	11.9	11.7
	Ave. Load on Flywheel Bearings - lbs.	41	41	42	44	45	42	41	43
NO.	TEMPERATURE DATA - DEG. F								
1	Test Pump Inlet	131	131	115	130	132	133	130	130
2	Test Pump Return	136	135	116	130	136	135	130	135
3	Test Pump Case Drain	198	208	210	212	209	204	213	200
4	Main Hyd. Reservoir Supply	123	124	120	125	125	123	125	120
5	Lube Oil Return	203	204	200	210	210	210	208	207
6	Lube Oil Suction	85	85	85	90	90	90	88	88
7	Scavange Oil Return	131	130	125	130	128	130	128	130
8	Gear Box Bearing Adjacent Pump	200	200	195	200	200	200	200	199
9	Gear Box Bearing Adjacent Flywheel	180	182	175	180	182	183	180	180
10	Inboard Flywheel Bearing	209	206	195	205	208	208	207	205
11	Outboard Flywheel Bearing	198	198	195	205	207	205	205	200
12	Flywheel Case Housing	170	170	160	173	178	176	175	175
NO.	SYSTEM PRESSURE DATA - PSIG								
13	Outboard Flywheel Bearing	28	28	23	24	24	24	24	24
14	Inboard Flywheel Bearing	28	28	23	23	25	26	26	26
15	Lube Pump Pressure Port	68	68	63	63	65	60	60	60
16	Lube Reservoir	0	0	0	0	0	0	0	0
17	Hyd. Supply Pump Press. Port	2150	2150	2150	2150	2150	2150	2150	2150
18	Test Motor Press. Port	2000	2000	2000	2000	2000	2000	2000	2000
19	Test Motor Return Port	110	110	110	110	110	110	110	110
20	Test Motor Case Drain	4	4	4	4	4	4	4	4
21	Vacuum at Flywheel Hub [illegible] - in. Hg	29.4	29.4	29.0	29.0	28.9	28.7	28.8	28.7
22	(Hub to Tip ΔP) Vacuum Across Flywheel Housing - in.Hg	1.4	1.4	0.8	0.8	0.8	0.7	0.8	0.7
23	Press at Flywheel Hub [illegible] - cm Hg Ab	2.3	2.3	3.5	3.4	3.4	3.6	3.5	3.5
NO.	FLUID FLOW DATA - GPM								
24	Inlet Flow to Motor	23	23	23	23	23	23	23	23
25	Case Drain Flow of Motor	.2-.4	.2-.4	.2-.4	.2-.4	.2-.4	.2-.4	.2-.4	.2-.4
26	Outboard Bearing Lube Flow cc/min	410	410	380	390	390	390	390	390
27	Inboard Bearing Lube Flow cc/min	410	410	380	395	405	405	405	450

*Start of a new days testing.

Table VII (Continued)

TILT TABLE SIMULATOR TEST DATA

Summary Data Sheet No. 5 Date: 3-1-67

No.	Item		16.5	17.0	17.5	18.0	18.5	19.0	19.5	20.0
	Running Time of Test - Hours		16.5	17.0	17.5	18.0	18.5	19.0	19.5	20.0
	Vibration Amp. at Flywheel Bearings-g's	Inboard	-	-	-	-	-	-	-	-
		Outboard	-	-	-	-	-	-	-	-
	(1) Flywheel Speed at Full Motoring-RPM		52,000	52,000	52,000	52,000	52,000	52,000	52,000	52,000
	(2) Minimum Speed Attain Under Load-RPM		44,500	44,500	44,500	44,500	44,500	44,500	44,500	44,500
	Total Precessional Load Cycles		6950	7152	7363	7575	7792	8005	8224	8440
	Ave. Tilt Table Velocity - deg/sec.		11.6	11.7	11.5	11.7	11.9	11.8	11.8	11.5
	Ave. Load on Flywheel Bearings - lbs.		42	43	42	43	41	42	41	42
NO.	TEMPERATURE DATA - DEG. F									
1	Test Pump Inlet		133	130	132	130	134	133	132	133
2	Test Pump Return		135	130	135	135	135	137	133	137
3	Test Pump Case Drain		202	215	203	212	215	210	215	212
4	Main Hyd. Reservoir Supply		123	125	125	125	126	127	126	127
5	Lube Oil Return		203	203	203	202	203	203	202	203
6	Lube Oil Suction		88	88	88	88	88	88	88	88
7	Scavange Oil Return		128	128	128	128	130	130	130	128
8	Gear Box Bearing Adjacent Pump		197	197	197	196	196	200	197	198
9	Gear Box Bearing Adjacent Flywheel		180	180	180	180	180	180	180	180
10	Inboard Flywheel Bearing		200	202	198	200	203	200	202	200
11	Outboard Flywheel Bearing		198	200	197	195	198	197	197	198
12	Flywheel Case Housing		175	170	170	170	170	170	170	168
NO.	SYSTEM PRESSURE DATA - PSIG									
13	Outboard Flywheel Bearing		25	25	24	26	26	25	27	26
14	Inboard Flywheel Bearing		26	25	24	25	25	24	26	25
15	Lube Pump Pressure Port		66	65	66	63	67	62	65	63
16	Lube Reservoir		0	0	0	0	0	0	0	0
17	Hyd. Supply Pump Press. Port		2150	2150	2150	2150	2150	2150	2150	2150
18	Test Motor Press. Port		2000	2000	2000	2000	2000	2000	2000	2000
19	Test Motor Return Port		110	110	110	115	115	115	115	115
20	Test Motor Case Drain		4	4	4	4	4	4	4	4
21	Vacuum at Flywheel Hub	- in. Hg	28.9	28.9	28.9	28.9	28.9	28.9	28.9	28.8
22	Vacuum Across Flywheel Housing (Hub to Tip ΔP)	- in.Hg	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.8
23	Press at Flywheel Hub	- cm Hg Ab	3.1	3.1	3.1	3.1	3.1	3.1	3.1	3.2
NO.	FLUID FLOW DATA - GPM									
24	Inlet Flow to Motor		23	23	23	23	23	23	23	23
25	Case Drain Flow of Motor		.2-.5	.2-.5	.2-.5	.2-.5	.2-.5	.2-.5	.2-.5	.2-.5
26	Outboard Bearing Lube Flow	cc/min	295	395	390	405	405	395	410	405
27	Inboard Bearing Lube Flow	cc/min	405	395	390	395	395	390	405	395

TILT TABLE SIMULATOR TEST DATA

Summary Data Sheet No. 6 Date: 3-1-67

No.	Item								
	Running Time of Test - Hours	20.5	21.0	21.5	22.0	22.5	23.0	23.5	24.0
	Vibration Amp. at Flywheel Bearings-g's — Inboard	-	-	-	-	-	-	-	-
	Vibration Amp. at Flywheel Bearings-g's — Outboard	-	-	-	-	-	-	-	-
	(1) Flywheel Speed at Full Motoring-RPM	52,000	52,000	52,000	52,000	52,000	52,000	52,000	52,000
	(2) Minimum Speed Attain Under Load-RPM	44,500	44,500	44,500	44,500	44,500	44,500	44,500	44,500
	Total Precessional Load Cycles	8676	8866	9088	9302	9516	9733	9950	10167
	Ave. Tilt Table Velocity - deg/sec.	11.7	11.8	12.1	11.9	12.0	12.1	11.8	11.6
	Ave. Load on Flywheel Bearings - lbs.	42	44	42	43	42	42	42	43
NO.	TEMPERATURE DATA - DEG. F								
1	Test Pump Inlet	135	134	135	135	135	135	137	130
2	Test Pump Return	136	138	138	140	138	138	138	135
3	Test Pump Case Drain	215	215	215	215	209	210	215	207
4	Main Hyd. Reservoir Supply	130	128	130	133	128	132	134	125
5	Lube Oil Return	205	210	205	208	208	210	206	200
6	Lube Oil Suction	88	90	90	88	88	90	90	88
7	Scavange Oil Return	130	128	130	128	128	127	130	126
8	Gear Box Bearing Adjacent Pump	200	205	202	204	204	202	203	190
9	Gear Box Bearing Adjacent Flywheel	180	184	185	183	180	180	183	175
10	Inboard Flywheel Bearing	202	210	210	208	206	208	207	194
11	Outboard Flywheel Bearing	198	205	205	204	202	205	205	194
12	Flywheel Case Housing	170	170	170	169	168	170	170	162
NO.	SYSTEM PRESSURE DATA - PSIG								
13	Outboard Flywheel Bearing	26	27	28	26	26	26	24	24
14	Inboard Flywheel Bearing	25	26	27	25	23	25	23	23
15	Lube Pump Pressure Port	65	67	68	67	67	67	65	68
16	Lube Reservoir	0	0	0	0	0	0	0	0
17	Hyd. Supply Pump Press. Port	2150	2150	2150	2150	2150	2150	2150	2150
18	Test Motor Press. Port	2000	2000	2000	2000	2000	2000	2000	2000
19	Test Motor Return Port	115	115	115	115	115	115	115	115
20	Test Motor Case Drain	4	4	4	4	4	4	4	4
21	Vacuum at Flywheel Hub - in. Hg	28.8	28.9	28.8	29.0	28.9	28.9	29.0	28.9
22	Vacuum Across Flywheel Housing - in.Hg (Hub to Tip ΔP)	0.8	0.8	0.8	0.8	0.8	0.8	0.8	0.8
23	Press at Flywheel Hub - mm Hg Ab	3.2	3.2	3.2	3.0	3.0	3.0	3.0	3.0
NO.	FLUID FLOW DATA - GPM								
24	Inlet Flow to Motor	23	23	23	23	23	23	23	23
25	Case Drain Flow of Motor	.2-.4	.2-.4	.2-.4	.2-.4	.2-.4	.2-.4	.2-.4	.2-.4
26	Outboard Bearing Lube Flow cc/min	405	410	420	405	405	405	385	385
27	Inboard Bearing Lube Flow	395	405	405	395	360	395	375	375

Table VII (Continued)

TILT TABLE SIMULATOR TEST DATA

Summary Data Sheet No. 7 Date: 3-2-67

		*24.5	25.0	25.5	26.0	26.5	27.0	27.5	28.0
Running Time of Test - Hours		*24.5	25.0	25.5	26.0	26.5	27.0	27.5	28.0
Vibration Amp. at Flywheel Bearings g's	Inboard	-	-	-	-	-	-	-	-
	Outboard	-	-	-	-	-	-	-	-
(1) Flywheel Speed at Full Motoring-RPM (2) Minimum Speed Attain Under Load-RPM		52,000	52,000	52,000	52,000	52,000	52,000	52,000	52,000
Total Precessional Load Cycles		10330	10550	10802	11057	11310	11330	11808	12060
Ave. Tilt Table Velocity - deg/sec.		14.4	14.1	14.1	14.1	14.3	14.1	14.0	13.6
Ave. Load on Flywheel Bearings - lbs.		62	60	61	60	60	59	59	58
NO.	TEMPERATURE DATA - DEG. F								
1	Test Pump Inlet	118	126	123	128	127	127	129	129
2	Test Pump Return	123	127	126	129	128	128	129	129
3	Test Pump Case Drain	196	200	200	197	200	200	198	198
4	Main Hyd. Reservoir Supply	111	115	120	116	115	115	115	115
5	Lube Oil Return	205	220	220	220	223	220	222	222
6	Lube Oil Suction	77	80	75	87	86	88	89	89
7	Scavange Oil Return	123	130	137	133	134	133	132	132
8	Gear Box Bearing Adjacent Pump	206	215	212	215	217	217	213	215
9	Gear Box Bearing Adjacent Flywheel	182	187	188	187	187	187	185	186
10	Inboard Flywheel Bearing	207	215	220	217	220	218	222	222
11	Outboard Flywheel Bearing	200	215	213	214	215	214	212	212
12	Flywheel Case Housing	157	170	180	180	180	180	180	179
NO.	SYSTEM PRESSURE DATA - PSIG								
13	Outboard Flywheel Bearing	19	26	26	26	26	26	26	26
14	Inboard Flywheel Bearing	19	26	26	26	27	27	27	27
15	Lube Pump Pressure Port	61	59	60	59	59	60	59	59
16	Lube Reservoir	0	0	0	0	0	0	0	0
17	Hyd. Supply Pump Press. Port	2000	1990	1990	1990	1990	1990	1990	1990
18	Test Motor Press. Port	2000	1990	1990	1990	1990	1990	1990	1990
19	Test Motor Return Port	120	120	120	120	115	115	115	115
20	Test Motor Case Drain	2	2	2	2	2	2	2	2
21	Vacuum at Flywheel Hub - in. Hg	29.1	29.1	29.0	28.9	29.0	29.0	28.9	29.0
22	(Hub to Tip ΔP) Vacuum Across Flywheel Housing - in.Hg	0.6	0.6	0.6	0.5	0.6	0.6	0.5	0.6
23	Press at Flywheel Hub - cm Hg Ab	2.3	2.5	2.7	2.8	2.6	2.6	2.7	2.7
NO.	FLUID FLOW DATA - GPM								
24	Inlet Flow to Motor	23	23	23	23	23	23	23	23
25	Case Drain Flow of Motor	.2	.2	.2	.2	.2	.2	.2	.2
26	Outboard Bearing Lube Flow cc/min	340	405	405	405	405	405	405	405
27	Inboard Bearing Lube Flow cc/min	340	405	405	405	415	415	415	415

Table VII (Continued)

TILT TABLE SIMULATOR TEST DATA

Summary Data Sheet No. 8 Date: 3-2-67

No.	Item								
	Running Time of Test - Hours	28.5	29.0	29.5	30.0	30.5	31.0	31.5	32.0
	Vibration Amp. at Flywheel Bearings-g's — Inboard	-	-	-	-	-	-	-	-
	Vibration Amp. at Flywheel Bearings-g's — Outboard	-	-	-	-	-	-	-	-
	(1) Flywheel Speed at Full Motoring-RPM	52,000	52,000	52,000	52,000	52,000	52,000	52,000	52,000
	(2) Minimum Speed Attain Under Load-RPM								
	Total Precessional Load Cycles	12320	12550	12800	13050	13300	13560	13800	14060
	Ave. Tilt Table Velocity - deg/sec.	13.9	13.8	13.7	14.3	14.2	14.1	14.0	14.1
	Ave. Load on Flywheel Bearings - lbs.	59	58	57	61	60	60	61	60
NO.	TEMPERATURE DATA - DEG. F								
1	Test Pump Inlet	129	130	130	127	127	128	128	128
2	Test Pump Return	129	131	132	128	127	129	129	129
3	Test Pump Case Drain	198	198	200	198	197	198	200	198
4	Main Hyd. Reservoir Supply	118	118	117	118	118	118	116	117
5	Lube Oil Return	222	217	216	218	212	212	210	209
6	Lube Oil Suction	90	83**	75	65	60	58	58	54
7	Scavenge Oil Return	134	131	130	127	128	128	129	129
8	Gear Box Bearing Adjacent Pump	215	211	210	210	207	205	208	205
9	Gear Box Bearing Adjacent Flywheel	186	188	185	182	183	181	180	180
10	Inboard Flywheel Bearing	218	210	210	210	208	206	209	209
11	Outboard Flywheel Bearing	213	209	207	206	205	202	201	201
12	Flywheel Case Housing	180	178	176	173	170	172	170	170
NO.	SYSTEM PRESSURE DATA - PSIG								
13	Outboard Flywheel Bearing	26	26	26	26	28	28	28	28
14	Inboard Flywheel Bearing	27	27	27	27	29	28	29	28
15	Lube Pump Pressure Port	59	59	59	59	59	59	60	60
16	Lube Reservoir	0	0	0	0	0	0	0	0
17	Hyd. Supply Pump Press. Port	1990	1990	1990	1990	1990	1990	1990	1990
18	Test Motor Press. Port	1990	1990	1990	1990	1990	1990	1990	1990
19	Test Motor Return Port	120	115	115	120	120	120	120	120
20	Test Motor Case Drain	2	2	2	2	2	2	2	2
21	Vacuum at Flywheel Hub - in. Hg	28.9	28.9	28.9	29.0	28.9	29.0	29.0	29.0
22	Vacuum Across Flywheel Housing - in.Hg (Hub to Tip ΔP)	0.5	0.5	0.5	0.5	0.4	0.4	0.4	0.4
23	Press at Flywheel Hub - cm Hg Ab	2.7	2.7	2.7	2.7	2.7	2.6	2.5	2.6
NO.	FLUID FLOW DATA - GPM								
24	Inlet Flow to Motor	23	23	23	23	23	23	23	23
25	Case Drain Flow of Motor	.2	.2	.2	.2	.2	.2	.2	.2
26	Outboard Bearing Lube Flow cc/min	405	405	405	405	420	420	420	420
27	Inboard Bearing Lube Flow cc/min	415	415	415	415	430	420	430	420

**Started to cool lube reservoir with liquid CO_2

Table VII (Concluded)

TILT TABLE SIMULATOR TEST DATA

Summary Data Sheet No. 9 Date: 3-2-67

No.	Item									
	Running Time of Test - Hours		32.5	33.0	33.5	*34.0	34.5			
	Vibration Amp. at Flywheel Bearings-g's	Inboard	-	-	4.2	4.5	4.5			
		Outboard	-	-	6.6	6.5	9.5			
	(1) Flywheel Speed at Full Motoring-RPM		52,000	52,000	52,000	53,000	53,000			
	(2) Minimum Speed Attain Under Load-RPM									
	Total Precessional Load Cycles		14300	14560	14780	15080	15280			
	Ave. Tilt Table Velocity - deg/sec.		14.1	14.2	14.0	14.0	14.0			
	Ave. Load on Flywheel Bearings - lbs.		60	62	64	64	-	(Failure of bearing)		
NO.	TEMPERATURE DATA - DEG. F									
1	Test Pump Inlet		128	128	125	132	138			
2	Test Pump Return		129	129	128	137	140			
3	Test Pump Case Drain		197	197	195	206	218			
4	Main Hyd. Reservoir Supply		118	118	118	126	142			
5	Lube Oil Return		207	220	218	220	295			
6	Lube Oil Suction		52	55	60	73	78			
7	Scavange Oil Return		123	131	128	126	118			
8	Gear Box Bearing Adjacent Pump		202	208	203	197	206			
9	Gear Box Bearing Adjacent Flywheel		181	181	183	178	184			
10	Inboard Flywheel Bearing		201	212	206	200	218			
11	Outboard Flywheel Bearing		200	210	206	208	296			
12	Flywheel Case Housing		169	171	170	146	225	to 310		
NO.	SYSTEM PRESSURE DATA - PSIG									
13	Outboard Flywheel Bearing		28	31	27	32	23			
14	Inboard Flywheel Bearing		28	29	28	33	24			
15	Lube Pump Pressure Port		60	60	60	70	70			
16	Lube Reservoir		0	0	0	0	0			
17	Hyd. Supply Pump Press. Port		1990	1990	1990	2860	2860			
18	Test Motor Press. Port		1990	1990	1990	2860	2860			
19	Test Motor Return Port		120	120	120	125	125			
20	Test Motor Case Drain		2	2	2	2	2			
21	Vacuum at Flywheel Hub - in. Hg		29.1	29.1	29.1	29.5	29.5			
22	Vacuum Across Flywheel Housing (Hub to Tip ΔP) - in.Hg		0.4	0.3	0.4	0.6	0.6			
23	Press at Flywheel Hub - cm. Hg Ab		2.4	2.4	2.4	1.3	1.3			
NO.	FLUID FLOW DATA - GPM									
24	Inlet Flow to Motor		23	23	23	23.6	23.6			
25	Case Drain Flow of Motor		0.2	0.2	0.2	0.2	0.2			
26	Outboard Bearing Lube Flow cc/min		420	430	410	460	375			
27	Inboard Bearing Lube Flow cc/min		420	430	420	470	385			

*Start new days testing.

Table VIII

TILT TABLE TEST RESULTS SUMMARY

a.	Flywheel, shaft moment of inertia	.266 in.-lb sec^2
b.	Flywheel, gearbox, motor moment of inertia	.268 in.-lb sec^2
c.	Upper flywheel speed	5,250 rad/sec
d.	Lower flywheel speed	4,660 rad/sec
e.	Average hydraulic motor speed	7,220 rpm
f.	Kinetic energy at upper speed	3.83 x 10^6 in.-lb
g.	Kinetic energy at lower speed	2.91 x 10^6 in.-lb
h.	Kinetic energy increase during motoring	0.92 x 10^6 in.-lb
i.	Time required to increase speed from lower speed to upper speed	10.0 sec
j.	Average power to accelerate	13.9 HP
k.	Angular acceleration of flywheel	59 rad/sec^2
m.	Torque at flywheel drive quill due to acceleration	15.72 in.-lb
n.	Hydraulic pressure drop across motor ports	1,890 lb/in.^{2}
o.	Displacement of hydraulic motor	.646 in.^{3}/rev
p.	Total power supplied to hydraulic motor due to displacement at average speed	22.2 HP
q.	Hydraulic motor shaft power at an assumed efficiency of 85 percent [(p) x .85]	18.9 HP
r.	Average gearbox and flywheel windage losses plus scavenge, lube pump and vacuum pump power extraction (q - j)	5.0 HP

TILT TABLE SIMULATOR TEST DATA RECORD

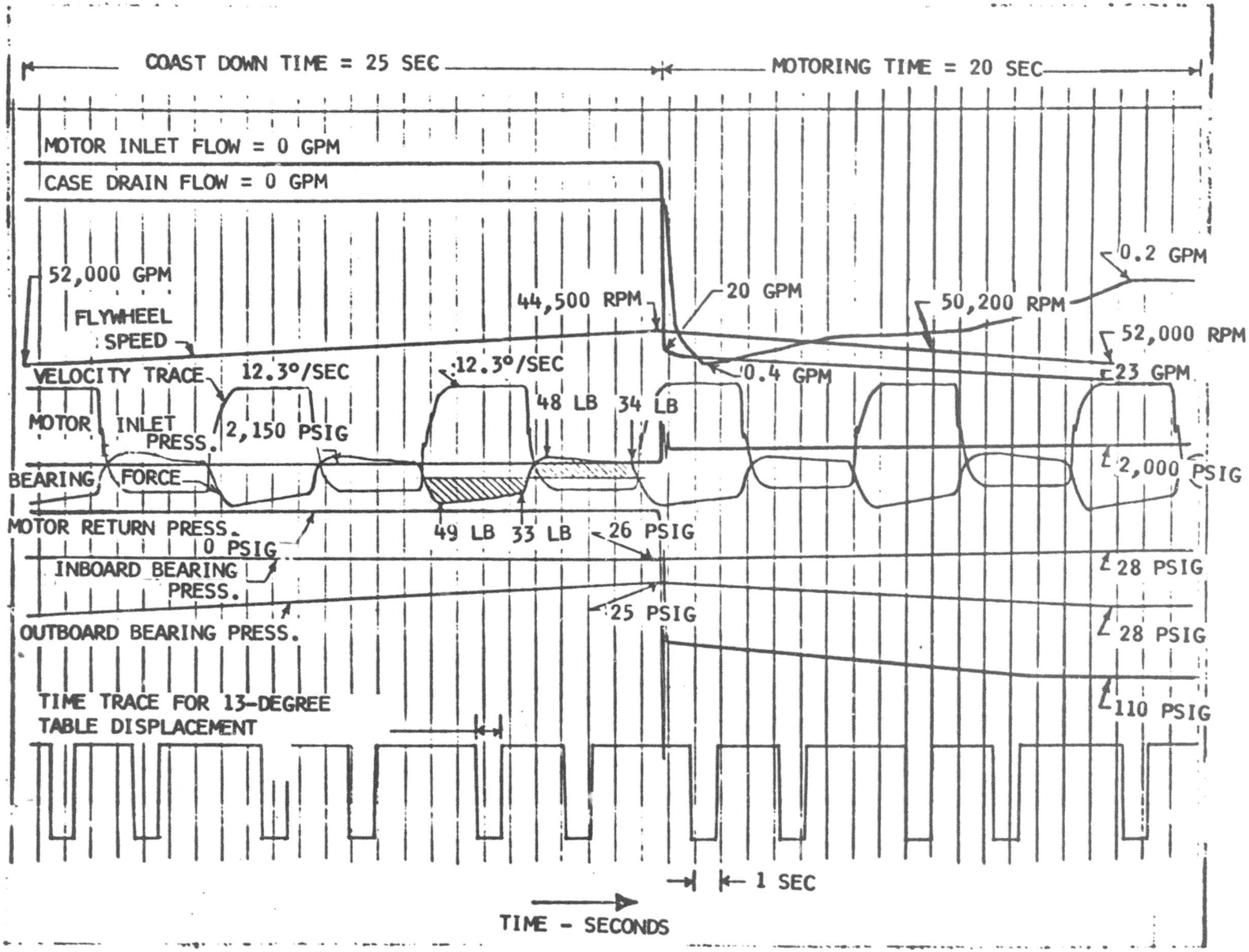


Figure 25. Tilt Table Test Oscillograph Record

Upon starting up for the next day's operation it was noticed that each time the flywheel tilted downward there was an audible screech. This was shortly followed by a runaway temperature rise on the outboard bearing and the test was shut down at 34.5 hours. Disassembly of the flywheel revealed that the outboard bearing (side farthest from the gearbox) had failed. The ball separator had completely disintegrated and had been carried away in the oil flow in the form of chips. The inner race was badly mutilated; however, all balls had remained in place. The outer race and balls had numerous cuts, scratches, and nicks but discoloration had not progressed to any great extent. The inner race was badly discolored and had, in fact, undergone plastic flow under the rolling load of the balls. No other parts of the flywheel had received any appreciable damage, except the oil slinger blades, which had been dented and bent by the pieces of the bearing separator which had passed through them. The disassembled flywheel is shown in figure 5.

The gearbox was returned to the supplier for teardown and inspection. All internal parts were deemed to be in good condition and reassembly did not require the replacement of any vital details.

Visual inspection of the surface of the flywheel revealed that carbon from the seals had discolored it somewhat all over; also, discoloration at the tip had occurred as a result of the combination of temperature plus the carbon and oil. The maximum temperature estimated at the tip from this discoloration was 400° F.

A basic objective of Tilt Table Testing was to determine as accurately as possible the actual flywheel losses as one of the major indices of performance. The following discussion outlines the method used to approximate the actual losses. In table VIII the summarized listing is based on the speed-up portion of the energy storage substation cycle. Two end conditions, i.e., conditions during lower flywheel speed and upper flywheel speed, can be satisfactorily defined from the measured data. Likewise, this speed-up time is established with fair accuracy.

Considering the complexity associated with demonstrated power extraction capability (in this case, indirectly measured by power input to the flywheel from the motor), it is impractical to treat the problem with other than average values. This stems from the fact that there are many variables introduced for which instantaneous values were not taken as data, and/or for which suitable empirical mathematical expressions are not available. Examples are: aerodynamic losses of the flywheel; windage losses in the gearbox; lube, scavenge, and vacuum pump power requirements; and pressure differential across the motor ports.

It is for these reasons that total energy was divided by total time to give average power. Average power was used to determine angular acceleration and shaft torque.

Power from the motor was then determined using 85 percent overall efficiency. This value is estimated and no attempt was made to establish it from previous engineering data on the motor, principally because this application uses a different hydraulic fluid (MIL-O-5606) from that which was used during its development (Oronite 8200).

Item r, table VIII (average gearbox and flywheel windage losses plus scavenge, lube, and vacuum pump power extraction) is, however, in the range of magnitude expected. A further breakdown of this loss can only be estimated. The flywheel had a differential pressure across it due to its own vacuum pumping effect. This does not lend itself readily to the power loss calculations shown in figure 3. Also, these losses are a function of speed to the 2-4/5 power. Taking these factors into account, it is believed that the flywheel losses of item r, table VIII, are less than 50 percent of the 5 horsepower.

Activity which followed on this simulator consisted of checkout of motor-pumps and flywheel-gearbox combinations. The flywheel-gearbox combinations are designated in order of their manufacture as energy storage substations No. 1, No. 2, and No. 3. The No. 1 energy storage substation was used in the above-mentioned endurance test.

When checkout runs of energy storage substation No. 2 were made, they proved to be not completely satisfactory in that higher bearing temperatures (220° to 235° F), and poorer flywheel housing vacuum were encountered, when compared to that experienced with the No. 1 energy storage substation.

The high bearing temperatures were believed to have resulted from one or both of two factors. The first is associated with a new seal which was installed on the hydraulic motor (XB-70 emergency generator motor). It had an improperly installed static seal which caused overheating, allowing heat to be conducted through the gearbox housing to the gearbox bearings. The second factor was the poorer performance of the flywheel vacuum system, which caused more than a normal amount of heat in the flywheel housing to be conducted to the flywheel bearings. The poorer vacuum was the result of the leaking flywheel shaft seal, and a leaking seal at the gearbox pump group housing. Upon replacement of these defective seals, the equipment operated with bearing temperatures around 170° F which were somewhat lower than those exhibited by the first assembly.

In the meantime, No. 1 energy storage substation had been rebuilt and was ready for checkout and continuation of the endurance test on the tilt table. However, checkout of this reassembled unit revealed higher bearing temperatures than it had exhibited upon initial assembly. It was believed that the possible cause was misalignment of the bearings which support the flywheel. For this reason, the flywheel assembly was disassembled and dimensionally checked. Alignment was found to be satisfactory; however, the lube jet ring O-rings were found to be "nibbled" and pieces of rubber were plugging the lubrication jets.

Since the disassembly and dimensional checks on this first assembly was known to be a time-consuming operation, it was decided to continue the tilt table endurance test using the No. 2 energy storage substation. This testing was scheduled to start with tilt rates equivalent to those called for by Test Objective II in table VI. The time scheduled for this tilt rate was 2.06 hours as an accelerated equivalent of Test Objective I and .45 hour for Test Objective II, plus an additional 2.05 hours to accomplish Test Objective III.

Investigation of the initial flywheel bearing failure led to a conclusion that it was premature and was probably the direct result of insufficient preload. A preload of 80 pounds had been intended; however, an inspection and measurement of detail parts disclosed that it was 69 pounds. Upon refurbishing the flywheel, the spring load was increased to 150 pounds.

Because this change was made, it was believed desirable to use the accelerated equivalent for Test Objective I. It was thought that the unit would quite easily pass through this segment of the test, and permit going on to segments of the test involving higher loads and their associated problems.

This test lasted for only 1 hour and 10 minutes. At this time the inboard bearing on the flywheel failed due to structural failure of the aluminum bronze ball separator. Inspection of the failed parts revealed that there was no evidence of overtemperature associated with this failure. It is believed that a short, sharp physical overload or series of overloads occurred on this separator and caused the failure. Although the exact nature of the mechanics of the overload application internal to the bearing is not known, it is quite clear that it resulted from the precessional loads associated with the high tilt rates.

It was also noted in connection with the higher tilt rates that the flywheel bearings would show a steady, slow temperature rise to unacceptable levels (230° F) until tilt table operation was interrupted. Upon cessation of tilting, the bearings would cool by 10° to 15° F to a lower stabilized operating temperature. In recognition of the fact that aircraft do not barrel roll through the sky for hours on end, it was decided to periodically interrupt the tilt-table operation to more closely simulate actual air vehicle operation. However, possibly because of the damage already done to the bearing before this approach was adopted, the time to cool down became progressively longer and the time to heat up became progressively shorter until the previously described failure occurred.

The data from this test are shown in table IX (two pages). The pickup points for these data may be determined by comparing the data numbers from table IX with those on the system schematic figure 15.

The Tilt Table Simulator was next used to check out the operation of the No. 3 energy storage substation. This unit displayed bearing temperatures around 202° F when operated by the Vickers hydraulic motor (XB-70 emergency generator motor). While this chechout was in progress the first of the New York Airbrake motor-pumps with automatic pressure sensing was received. This unit was checked out on the same (No. 3) energy storage substation and a bearing stabilization temperature of 215° F was noted.

It is believed that the difference in stabilized bearing temperatures between the two hydraulic motors stems from the magnitude of the vibration introduced by the stroking of the pistons. The magnitude of the vibration, in turn, is a function of the operating pressure. In the case of the emergency generator motor a lower hydraulic pressure (2,000 versus 3,000 psi) is used resulting in lower piston stroke vibrational accelerations for that motor (3.5G versus 7G).

Table IX

TILT TABLE SIMULATOR TEST DATA
(SUBSTATION NO. 2)

Summary Data Sheet No. 1 Date: 4-5-67

NO	Item								
	Running Time of Test - Hours — Running	0.25	0.50	1.00	1.5	1.93	2.25	2.42	2.92
	Running Time of Test - Hours — Tilting	--	--	0.50	--*	0.80	--*	0.97	1.13
	Vibration Amp. at Flywheel Bearings-g's — Inboard	3.0	3.0	4.2	3.4	4.2	3.5	4.1	4.1
	Vibration Amp. at Flywheel Bearings-g's — Outboard	3.0	3.0	4.2	3.4	4.2	3.6	4.0	4.1
	(1) Flywheel Speed at Full Motoring-RPM	52,400	52,400	52,400	52,400	52,400	52,400	52,400	52,400
	(2) Minimum Speed Attain Under Load-RPM	--	--	--	--	--	--	--	--
	Total Precessional Load Cycles	--	--	465	--	750	--	974	1096
	Ave. Tilt Table Velocity - deg/sec.	--	--	30	--	30	--	30	35
	Ave. Load on Flywheel Bearings - lbs.	--	--	132	--	131	--	132	145
NO	TEMPERATURE DATA - DEG. F								
1	Test Pump Inlet	112	125	127	130	125	131	127	128
2	Test Pump Return	118	130	130	135	130	135	132	132
3	Test Pump Case Drain	192	200	200	200	200	202	204	198
4	Main Hyd. Reservoir Supply	105	115	120	122	122	122	120	123
5	Lube Oil Return	136	150	230	186	225	188	220	219
6	Lube Oil Suction	75	75	75	80	78	82	87	82
7	Scavange Oil Return	112	120	150	140	148	140	154	150
8	Gear Box Bearing Adjacent Pump	168	178	220	198	225	200	226	228
9	Gear Box Bearing Adjacent Flywheel	166	170	192	182	195	185	195	198
10	Inboard Flywheel Bearing	135	150	220	182	230	188	220	235
11	Outboard Flywheel Bearing	135	150	224	182	225	188	218	212
12	Flywheel Case Housing	115	137	200	170	195	168	190	185
NO	SYSTEM PRESSURE DATA - PSIG								
13	Outboard Flywheel Bearing	26	26	35	36	35	35	35	40
14	Inboard Flywheel Bearing	26	26	36	37	36	36	36	42
15	Lube Pump Pressure Port	57	55	50	52	50	50	50	50
16	Lube Reservoir	0	0	0	0	0	0	0	0
17	Hyd. Supply Pump Press. Port	2480	2480	2450	2450	2450	2450	2450	2450
18	Test Motor Press. Port	2340	2340	2340	2340	2340	2340	2340	2340
19	Test Motor Return Port	110	110	110	110	110	110	110	110
20	Test Motor Case Drain	4	4	4	4	4	4	4	4
21	Vacuum at Flywheel Hub - in. Hg	29.4	29.35	29.2	29.2	29.1	29.2	29.2	29.2
22	(HUB TO TIP ΔP) Vacuum Across Flywheel Housing - in.Hg	0.7	0.7	0.9	0.9	1.0	1.0	1.0	1.1
23	Pressure at Flywheel Hub - CM Hg Ab	2.5	2.6	3.5	3.1	3.6	2.9	3.0	3.1
NO	FLUID FLOW DATA - GPM								
24	Inlet Flow to Motor	23	23	23	23	23	23	23	23
25	Case Drain Flow to Motor	0.2	0.2	0.25	0.25	0.25	0.25	0.25	0.25
26	Outboard Bearing Lube Flow cc/min.	410	410	500	500	500	500	500	585
27	Inboard Bearing Lube Flow cc/min.	400	400	500	500	500	500	500	595

*Stopped precessional load cycling during this period to allow bearing temperatures to decrease.

Table IX (Concluded)

TILT TABLE SIMULATOR TEST DATA
(SUBSTATION NO. 2)

Summary Data Sheet No. 2 Date: 4-5-67

NO	Item		Value
	Running Time of Test - Hours	Running	3.25
		Tilting	1.17
	Vibration Amp. at Flywheel Bearings-g's	Inboard	4.5
		Outboard	4.2
	(1) Flywheel Speed at Full Motoring-RPM		52,400
	(2) Minimum Speed Attain Under Load-RPM		-
	Total Precessional Load Cycles		1124
	Ave. Tilt Table Velocity - deg/sec.		35
	Ave. Load on Flywheel Bearings - lbs.		147
NO	TEMPERATURE DATA - DEG. F		
1	Test Pump Inlet		130
2	Test Pump Return		135
3	Test Pump Case Drain		198
4	Main Hyd. Reservoir Supply		125
5	Lube Oil Return		204
6	Lube Oil Suction		83
7	Scavange Oil Return		145
8	Gear Box Bearing Adjacent Pump		227
9	Gear Box Bearing Adjacent Flywheel		190
10	Inboard Flywheel Bearing		250 ← Inboard bearing failed.
11	Outboard Flywheel Bearing		205
12	Flywheel Case Housing		175
NO	SYSTEM PRESSURE DATA - PSIG		
13	Outboard Flywheel Bearing		32
14	Inboard Flywheel Bearing		32
15	Lube Pump Pressure Port		50
16	Lube Reservoir		0
17	Hyd. Supply Pump Press. Port		2480
18	Test Motor Press. Port		2340
19	Test Motor Return Port		110
20	Test Motor Case Drain		4
21	Vacuum at Flywheel Hub - in. Hg		29.2
22	(HUB TO TIP ΔP) Vacuum Across Flywheel Housing - in.Hg		1.1
23	Pressure at Flywheel Hub - CM Hg Ab		3.1
NO	FLUID FLOW DATA - GPM		
24	Inlet Flow to Motor		23
25	Case Drain Flow to Motor		0.25
26	Outboard Bearing Lube Flow cc/min.		455
27	Inboard Bearing Lube Flow cc/min.		455

During these checkout operations of the motor-pump it was discovered that bronze particles were being generated and discharged from the case drain line. Operation of the motor-pump was immediately discontinued and it was returned to the supplier for inspection and refurbishment. A review of the test data records revealed that the return port was not being pressurized to a level as high as would be recommended by New York Airbrake. During portions of the motoring and pumping cycle the pressure would drop as low as 60 psi. (A minimum of 70 psi is recommended by the supplier.) While the motor-pump was being refurbished, the No. 3 energy storage substation was removed from the Tilt Table and installed on the Intermittent Duty Cycle Simulator.

After refurbishment the motor-pump unit was redelivered and it checked out satisfactorily on the Tilt Table Simulator using energy storage substation No. 1. The flow and pressure settings were adjusted to proper calibration. It was then installed on the Intermittent Duty Cycle Simulator.

The Tilt Table Simulator was next used to check out the No. 2 energy storage substation following its refurbishment due to the second endurance test failure. After a short period of operation, it was suspected from the temperature readings that leakage into the evacuated portion of the flywheel was occurring. To verify this, an external vacuum pump was attached to the drain line at the bottom of the flywheel housing. In approximately 2 minutes a fluid quantity approaching a quart was withdrawn while the flywheel was maintaining 52,000 rpm.

It is significant to observe that this excessive oil leakage into the flywheel cavity did not create dangerous temperature conditions (evacuated oil reached 270° F) or create destructive drag forces.

Later disassembly of the flywheel disclosed that improperly sized static O-ring seals had inadvertently been installed where the carbon face seal cartridge is sealed to the flywheel housing.

The precessional load endurance test was next attempted using the energy storage substation No. 1 and the Vickers emergency generator motor. Upon checkout of this test arrangement, an inadvertent closing off of the case drain line led to failure of the rotating group.

The test was resumed using the prototype New York Airbrake motor-pump, i.e., the one using manual control to effect motoring and pumping modes. Approximately 4-1/2 hours of test time was completed using a tilt table precessional rate of 11.7 degrees/second (54-pound bearing load) before the test was terminated due to impending failure of the motor-pump. This was indicated by the fact that a large amount of metal chips were being produced and discharged through the case return port. The data from this test are shown in table X (one page). Figure 26 is a sample data record from which some of the data were taken, and which were used to monitor performance.

Following termination of the third precessional load endurance test, the final two New York Airbrake motor-pumps with automatic pressure-sensing control were received. These units were checked out and calibrated with respect to pressure and flow (speed) on the tilt table. After checkout, one unit was installed on the Intermittent Duty Cycle Simulator as a replacement for the

Table X

TILT TABLE SIMULATOR TEST DATA
(SUBSTATION NO. 1)

SUMMARY DATA SHEET NO. 1 DATE: 5-22-67

NO.	Item								
	Running Time of Test - Hours	0.5	1.0	1.5	2.0	2.5	3.0	3.5	4.0
	Vibration Amp. at Flywheel Bearings - g's — Inboard	*	*	6.5	7.3	8.5	8.5	7.5	7.5
	Vibration Amp. at Flywheel Bearings - g's — Outboard	6.5	8.0	6.5	9.0	9.7	9.7	9.7	9.1
	(1) Flywheel Speed at Full Motoring-RPM	52,000	52,000	52,000	52,000	52,000	52,000	52,000	52,000
	(2) Minimum Speed Attain Under Load-RPM	38,000	38,000	38,000	38,000	38,000	38,000	38,000	38,000
	Total Precessional Load Cycles	84	210	334	562	880	1150	1410	1690
	Ave. Tilt Table Velocity - deg/sec.	11.7	11.7	11.7	11.7	11.7	11.7	11.7	11.7
	Ave. Load on Flywheel Bearings - lbs.	--	49-57	51-52	48-59	49-58	47-56	48-59	49-59
NO.	TEMPERATURE DATA - DEG. F								
1	Test Pump Inlet	110	125	130	135	134	126	130	135
2	Test Pump Return	125	128	128	119	125	125	127	130
3	Test Pump Case Drain	178	178	178	166	170	172	175	172
4	Main Hyd. Reservoir Supply	123	125	126	115	122	124	125	127
5	Lube Oil Return	210	215	214	195	205	210	207	210
6	Lube Oil Suction	78	80	81	75	80	83	85	87
7	Scavenge Oil Return	160	160	162	150	150	155	155	155
8	Gear Box Bearing Adjacent Pump	200	192	194	180	185	190	188	190
9	Gear Box Bearing Adjacent Flywheel	210	202	204	185	195	195	196	197
10	Inboard Flywheel Bearing	200	202	205	184	195	195	196	197
11	Outboard Flywheel Bearing	198	202	205	185	195	195	196	197
12	Flywheel Case Housing	162	165	170	147	165	170	160	160
NO.	SYSTEM PRESSURE DATA - PSIG								
13	Outboard Flywheel Bearing	32	30	30	30	30	30	30	30
14	Inboard Flywheel Bearing	34	31	31	31	31	31	31	31
15	Lube Pump Pressure Port	50	50	50	51	50	50	50	50
16	Lube Reservoir	0	0	0	0	0	0	0	0
17	Hyd. Supply Pump Press. Port	3360	3360	3360	3360	3275	3275	3275	3275
18	Test Motor Press. Port (Motoring)	3340	3340	3340	3340	3200	3200	3250	3250
19	Test Motor Return Port (Motoring)	120	120	120	120	120	120	120	120
20	Test Pump Press. Range (Pumping)	2850 2300	2850 2300	2850 2300	2850 2300	2850 2300	2850 2300	2850 2300	2850 2300
21	Test Pump Suction Press. (Pumping)	125	125	125	125	125	125	125	125
22	Test Motor - Pump Case Drain	4	4	4	4	4	4	4	4
23	Vacuum at Flywheel Hub - in. Hg.	29.5	29.4	29.4	29.2	29.0	29.0	29.0	29.0
24	(HUB TO TIP ΔP) Vacuum Across Flywheel Housing-in. Hg.	0.75	0.75	0.75	-	-	-	-	-
25	Pressure at Flywheel Hub - CM Hg Ab.	1.7	1.7	1.7	2.0	2.6	2.6	2.6	2.6
NO.	FLUID FLOW DATA - GPM								
26	Inlet Flow to Motor	13.5	13.5	15.3	15.0	15.2	15.1	15.1	15.1
27	Motor-Pump Case Drain Flow	.5-.7	.5-.7	.5-.7	.5-.7	.5-.7	.5-.7	.5-.7	.5-.7
28	A.P.U. Flow Rate	18.5	18.5	18.5	18.5	18.5	18.5	18.5	18.5
29	Combined A.P.U. & Test Pump Flow	32.0 30.0	32.0 30.0	32.0 29.2	31.6 29.2	31.6 29.2	31.6 29.2	31.6 29.2	31.6 29.2
30	Test Pump Flow Rate	13.5 11.5	13.5 11.5	13.5 10.7	13.1 10.7	13.1 10.7	13.1 10.7	13.1 10.7	13.1 10.7
31	Outboard Bearing Lube Flow - cc/min.	775	730	730	730	730	730	730	730
32	Inboard Bearing Lube Flow - cc/min.	800	760	760	760	760	760	760	760

*Bad data, replaced transducer.

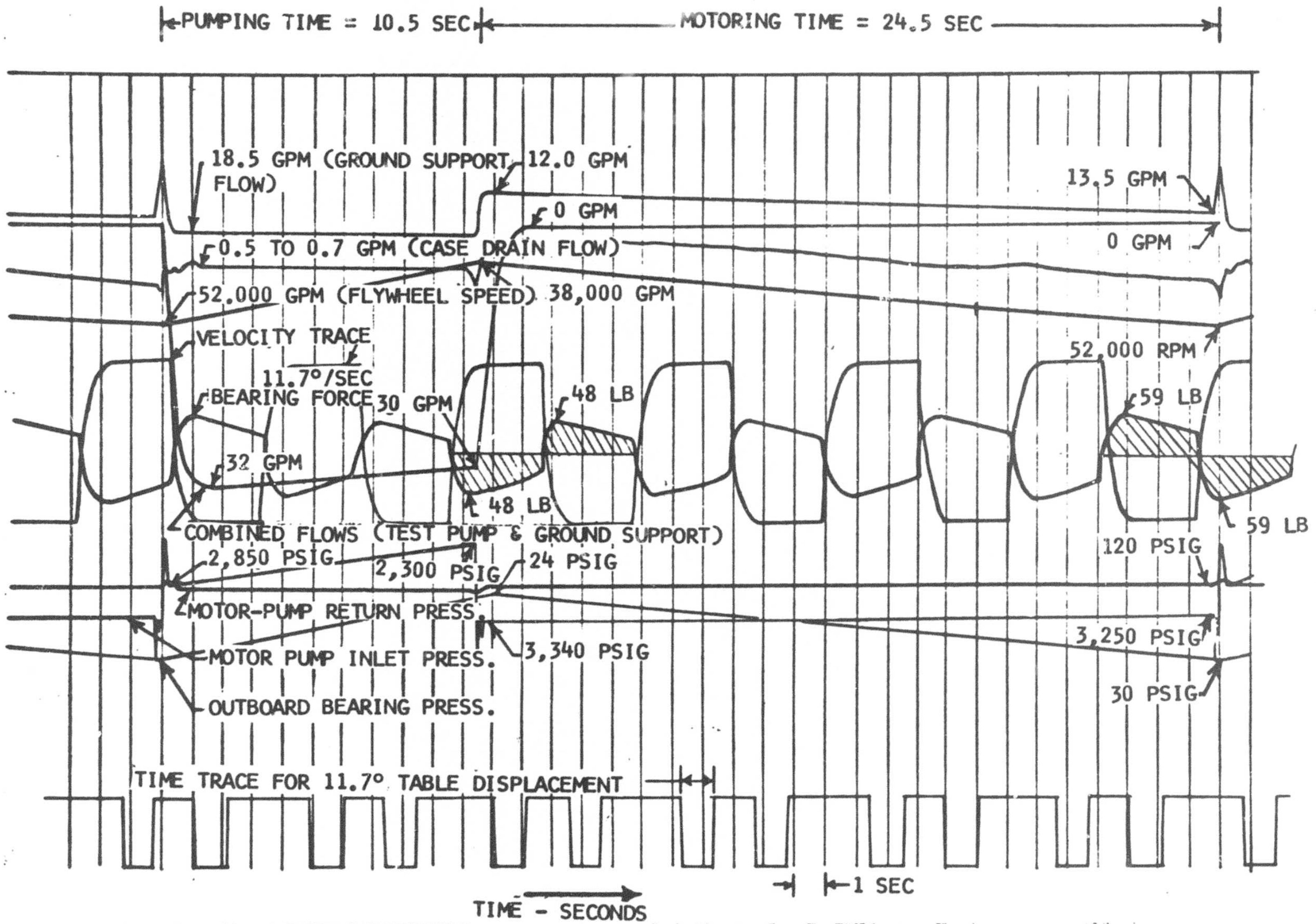


Figure 26. Tilt Table Test Oscillograph Record

first (automatic motor-pump) delivered unit which failed and was returned to the supplier for refurbishment. The other motor-pump was retained at the tilt table for use in the Continuous Duty Cycle Simulator. Tilt Table Simulator testing was discontinued at this point. The Continuous Duty Cycle Simulator was planned to utilize portions of the Tilt Table Simulator. The schedule requirements associated with its fabrication did not permit further exploring of the bearing problems as a Tilt Table.

Prior to operation of the simulator, an analysis of the possible resonant conditions of vibration was made on an IBM 7094 computer using a cathode ray tube graph-plotting camera for output results, as well as numerical printout. The FORTRAN program which was employed took all factors except one into careful consideration. The physical factors such as size, shape, inertia, bearing spring rates, etc, were quite adequately estimated. However, damping could not be adequately estimated and was neglected in the computation. Figure 27 is a schematic of the spring system used as a basis for the computer program.

The computer analysis predicted serious problems as indicated in table XI. However, as stated in the foregoing test description, the vibration search ". . . revealed no resonance conditions of measurable magnitude that could be positively identified. . ."

Configuration "A" in table XI was intended to most closely represent the configuration as tested on the Tilt Table. It should be noted that K_2, K_3, etc, are the rates for the springs indicated on the system schematic of figure 27. Configurations "B" through "G" are other spring system arrangements which were investigated in case actual testing indicated a change in design was in order.

D. TILT TABLE SIMULATOR TEST CONCLUSIONS

As a result of the Tilt Table Simulator testing, a high confidence level can be placed on the feasibility of successfully designing energy storage flywheels for airborne applications. The amount of precessional loading endured by the test article would qualify it for installation, with selective position orientation, in aircraft with pitching characteristics similar to heavy long-range bombers and supersonic transports for periods in excess of 3,000 flight hours. The test results would also indicate that this unit, as tested, could be selectively orientated in fighter reconnaissance and low-level strategic-type aircraft and operated without servicing for periods up to 1,000 hours. In each of these cases "selective orientation" means mounting with flywheel axis parallel to the fuselage centerline such that the flywheel bearings would not feel "roll" induced precessional loads.

There are many improvements which can be made to the bearing design, and it would, on this basis, be possible to lengthen the time between replacements and/or increase the flexibility of position orientation.

The failures experienced indicate that the bearing ball fit-up in the separator could probably be improved and separator strength increased, thus lengthening the separator life to a point where race subsurface fatigue became the limiting factor. In that connection even further improvement could

Table XI

COMPUTED BEARING FORCES

	Critical Speeds		Bearing Force at Upper Critical at $\alpha = 720$ Rad/Sec^2	Bearing Force During Decceleration at $\alpha = 15.4$ Rad/Sec^2
	Vertical Mode	Horizontal Mode		
Present				
Configuration "A"	25,821	766		
KBRG	27,878	789	1,200 lb	8,250 lb
K_2 = 215,000 psi	36,143	34,696		
K_3 = 31,000 psi				
Configuration "B"				
KBRG	788			
K_2 = 1,000 psi	862			
K_3 = 1,000 psi	35,831			
Configuration "C"				
K_1 = 10,000 psi	6,059	765		
K_2 = 100,000 psi	9,894	789	60 to 70 lb	410 to 478
K_3 = 3,115 psi	10,456	6,683		
Configuration "D"				
K_1 = 10,000 psi	6,337			
K_2 = 200,000 psi	13,904			
K_3 = 200,000 psi	14,380			
Configuration "E"				
K_1 = 10,000 psi	6,440			
K_2 = 400,000 psi	19,567			
K_3 = 400,000 psi	19,663			
Configuration "F"				
K_1 = 20,000 psi	6,871			
K_2 = 75,000 psi	8,554		37 lb	253 lb
K_3 = 75,000 psi	11,252			
Configuration "G"				
K_1 = 40,000 psi	7,305			
K_2 = 75,000 psi	8,663		85 lb	582 lb
K_3 = 75,000 psi	14,861			

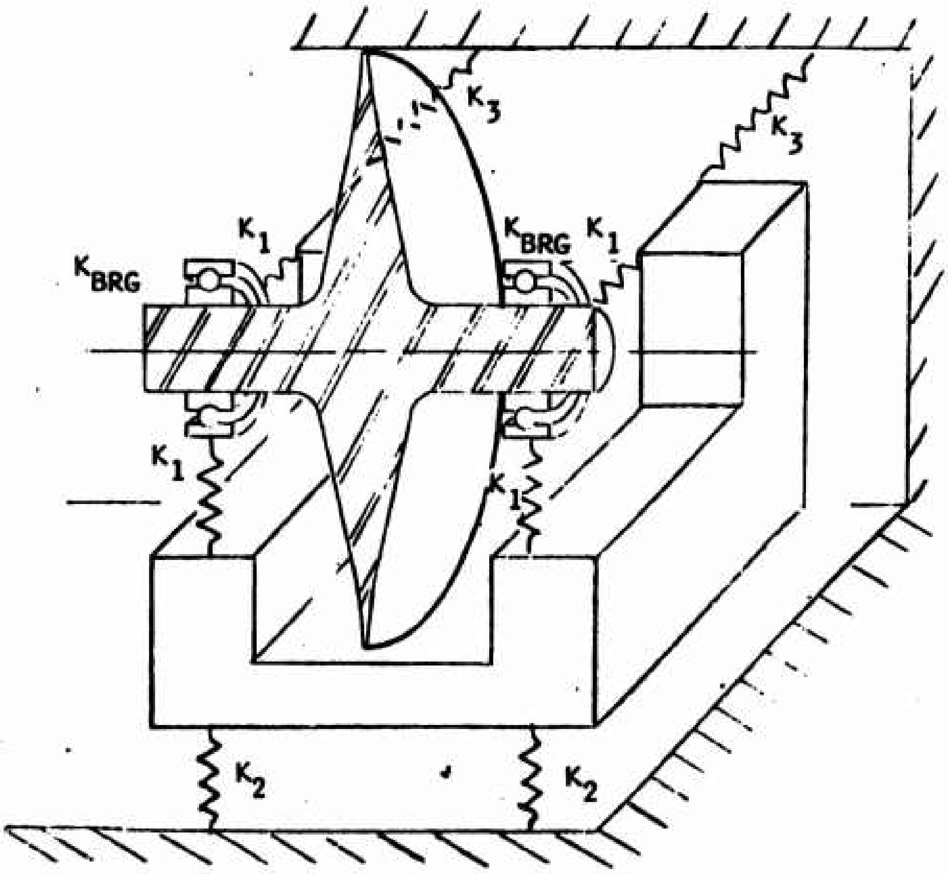


Figure 27. Spring System Schematic

be made by the use of improved steel, such as M-50 tool steel. In addition to improved fatigue life, the M-50 races would allow higher preload, thus reducing the ball path eccentricity, which causes the major loading on the separator.

In addition to using improved bearings, the problem of meeting precessional loads can be ameliorated through redesign of the flywheel. This redesign would involve a smaller diameter, higher speed flywheel of "chunkier" cross section.

A reduction in diameter reduces the precessional loads proportionately. However, a reduction in diameter also increases the base width of the optimized flywheel such that the bearings can be spaced wider apart without changing the resonant vibration conditions. Thus, decreasing the diameter of the flywheel effectively reduces the precessional loads by approximately a factor of two. In addition, it reduces the windage losses by the 1.8 power of the reduction in radius.

The only disadvantage of decreasing the diameter of the optimized flywheel is that it causes the speed to increase in the same proportion. This increase in speed increases the operating DN value of the support bearings, unless the bore diameter is also decreased. Each individual application must be analyzed for its own operating condition; however, it can be generalized that as the speed increases, the torque in the shaft decreases. This would suggest that the DN value for the bearings can usually be held down to acceptable limits regardless of the diameter of the flywheel, until the speed exceeds the practical limits of gearing.

Tilt Table Simulator testing established that by using the best available balancing technology, there will be essentially no problems with resonant vibration conditions induced by the flywheel itself.

It is believed that the poor correlation between measured and computed vibration results indicates that computer results cannot be relied upon for future flywheel designs. This would suggest that resonance conditions induced by flywheel out of balance should be tested for early in the fabrication program of energy storage substations.

Another principle demonstrated by the Tilt Table Simulator is that it is feasible to use a single gear mesh to couple a hydraulic rotating group to a high speed flywheel.

The vacuum system used in the flywheel was demonstrated to be adequate even though it was of small capacity and was a gear pump design. It is believed that a near optimum size was used when its size is considered to be a trade-off between pump power consumption versus flywheel aerodynamic power loss.

Section IV

INTERMITTENT DUTY CYCLE SIMULATOR

A. BASIC TEST OBJECTIVES

The specific objective of the Intermittent Duty Cycle Simulator endurance tests was to demonstrate the use of a flywheel as a source of energy for accomplishing an intermittent actuation function. As a result of this demonstration and the data which were created, the feasibility and weight improvement of applying the flywheel energy storage concept to this type of aircraft actuation function could be evaluated. To this end the XB-70 landing gear extension and retraction cycles were selected. The nature of this duty cycle identifies it as "intermittent" in that it has short periods of peak demand followed by long quiscent periods of zero demand.

Two endurance tests were planned to demonstrate the flywheel's adaptability to utility (intermittent duty cycle) functions. Each of these demonstrations were to operate the simulated XB-70 landing gear through 10,000 cycles. The first demonstration employed a hydraulic motor-pump to supply energy to, and extract energy from, the flywheel. The second demonstration blocked off the pumping function and used the motor-pump solely as a motor to supply energy to the flywheel. Energy was extracted through the power takeoff (PTO) shaft on the gearbox and delivered to the simulated landing gear through a clutch-brake unit and a mechanical hinge.

Figures 28 and 29 are simplified schematic diagrams of the two test arrangements. Figure 30 is a detailed schematic diagram of the simulator showing both hydraulic and mechanical power extraction equipment.

B. HYDRAULIC POWER EXTRACTION SYSTEM AND ENDURANCE TEST

Hydraulic power extraction is accomplished by causing the motor-pump hydraulic unit to assume a pumping mode at the time, or shortly after the time, that gear up or gear down is selected. The conversion from pumping to motoring is accomplished by sensing the pressure drop in the supply line that occurs as the flow capacity of the main system pump is exceeded.

It will be noted that the control of the hydraulic actuator stroke versus time is accomplished by a system employing electrically controlled servo valves. This arrangement was selected principally because the equipment and the technology to successfully implement and maintain it throughout the 10,000 cycles of testing, was readily available in the Structures Laboratory where the Intermittent Duty Cycle Simulator (XB-70 Landing Gear Simulator) was located. This equipment has, in the past, been used to control the hydraulic actuators used to apply fatigue test loading to large aircraft structures.

Duty cycle control was accomplished through a stroke versus time duty cycle curve plotted on a constant speed rotating drum. A curve following device compared the plotted position of this curve with the position of the landing gear as supplied by a position transducer which was fastened

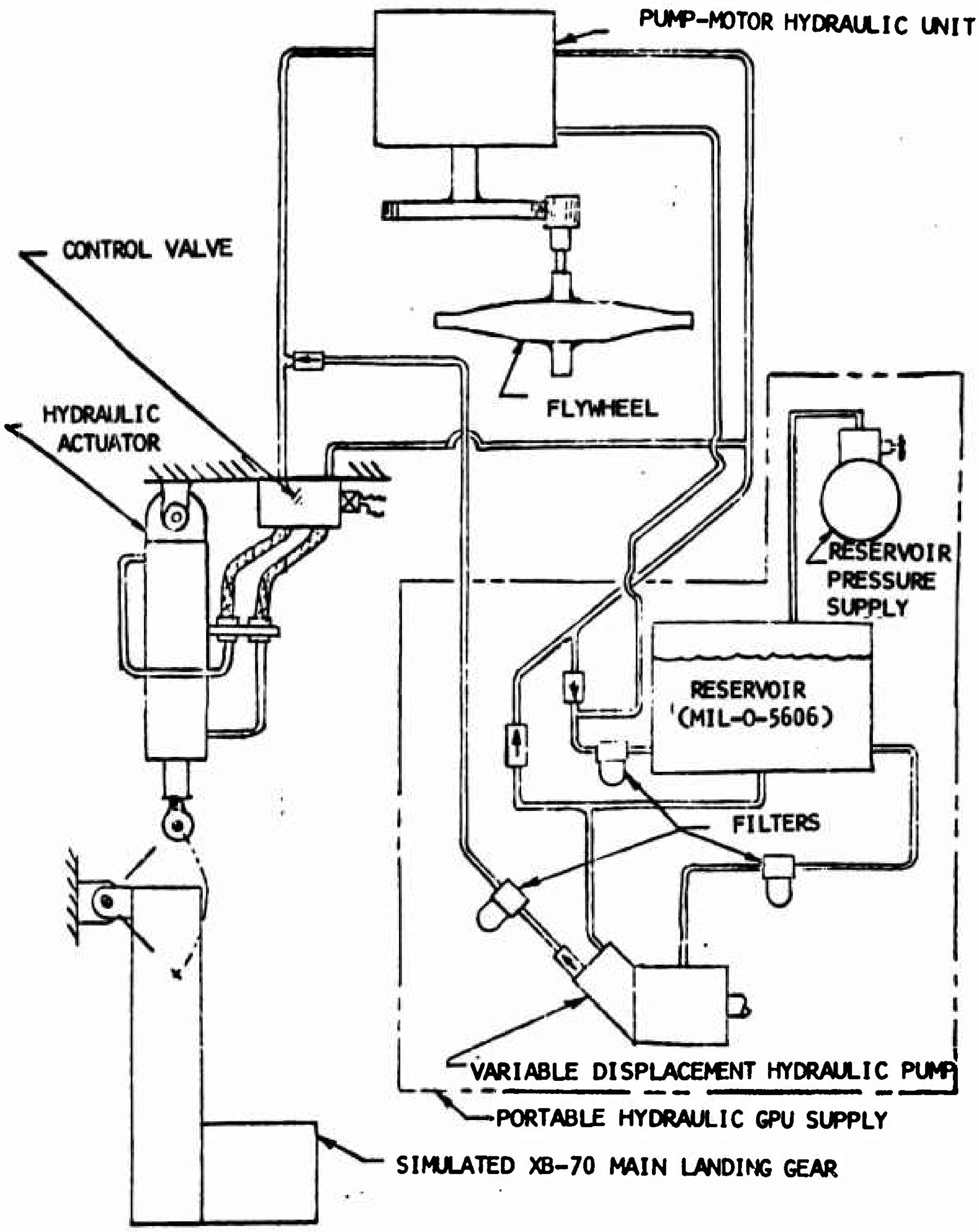


Figure 28. Hydraulic Power Extraction System

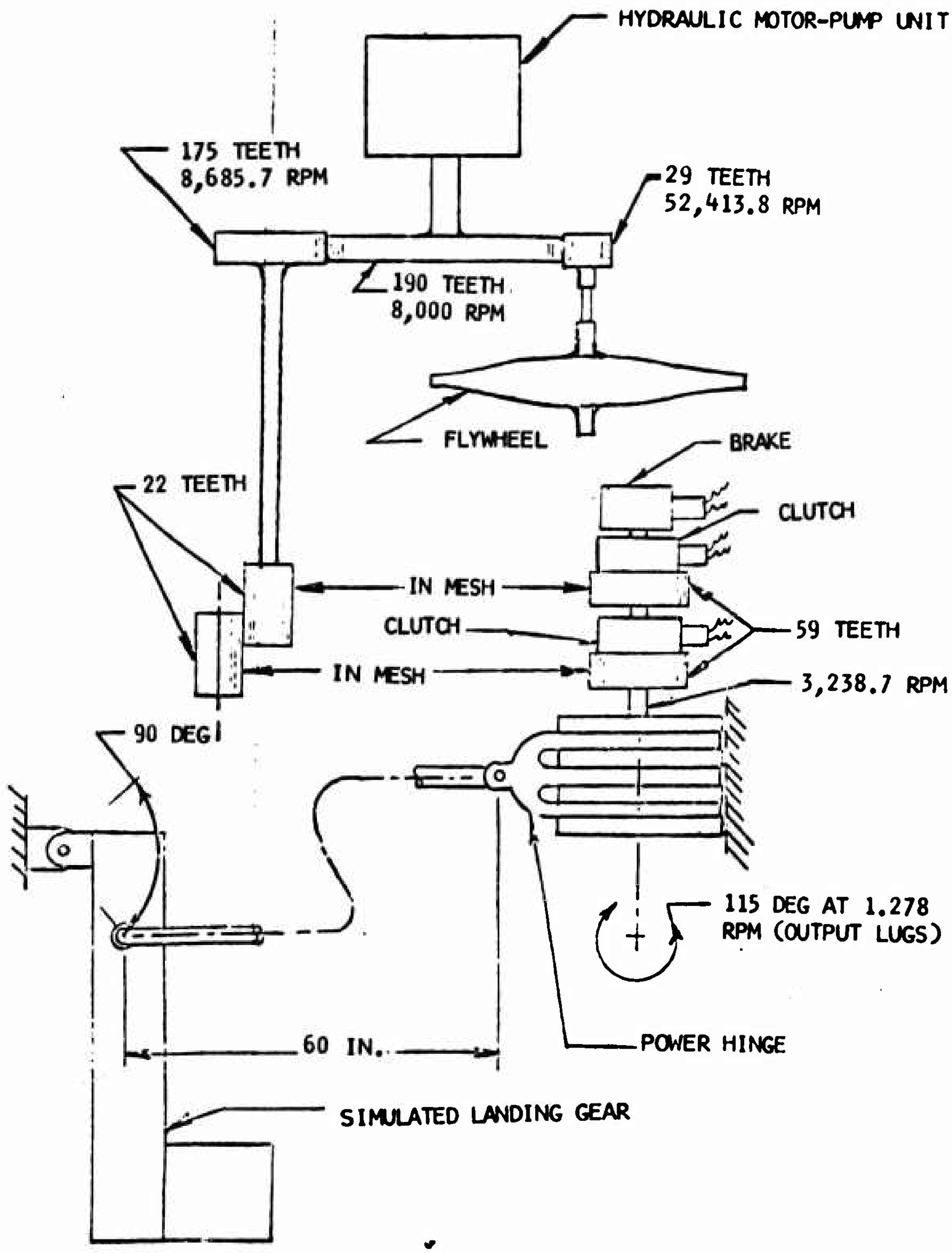
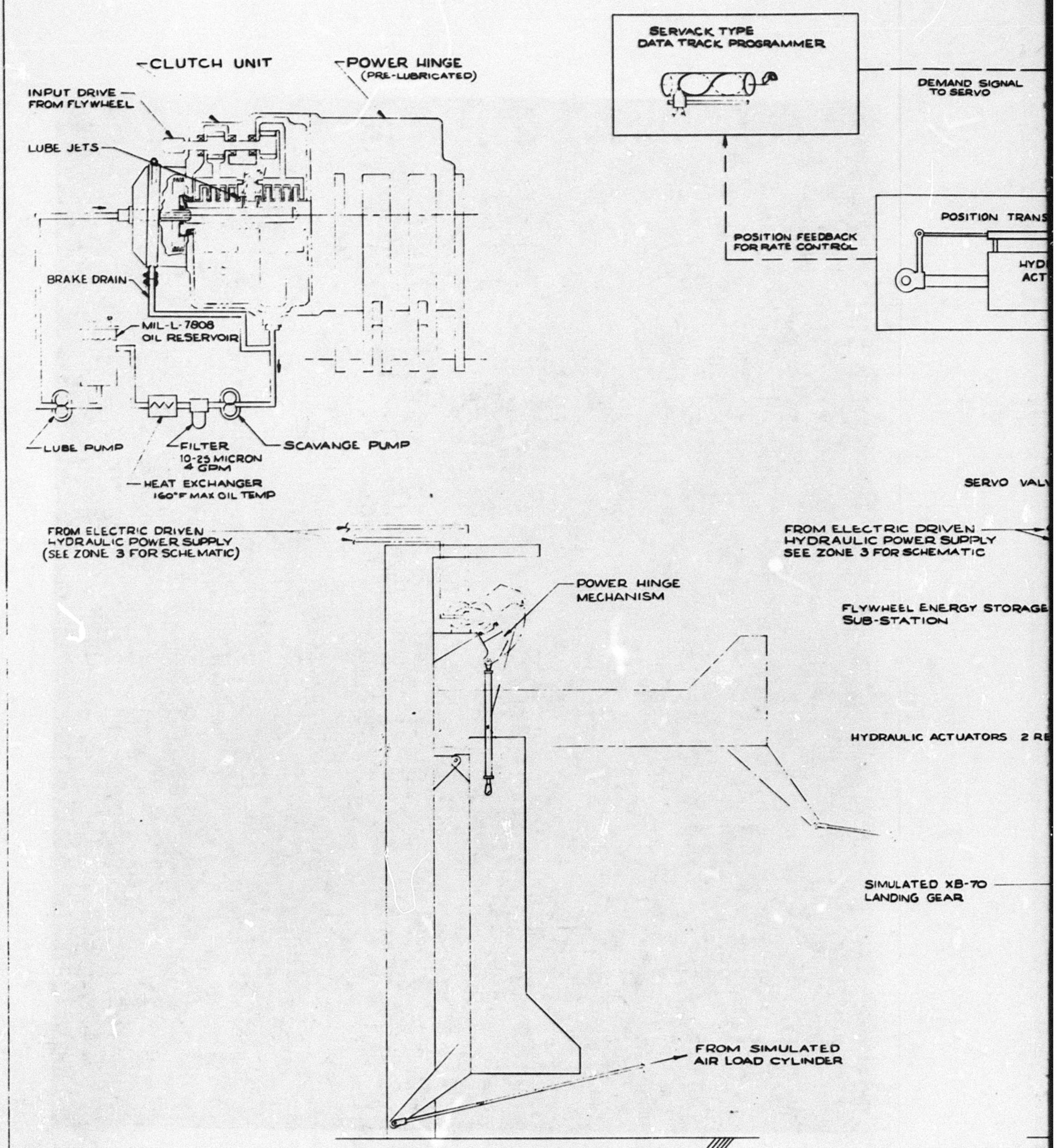


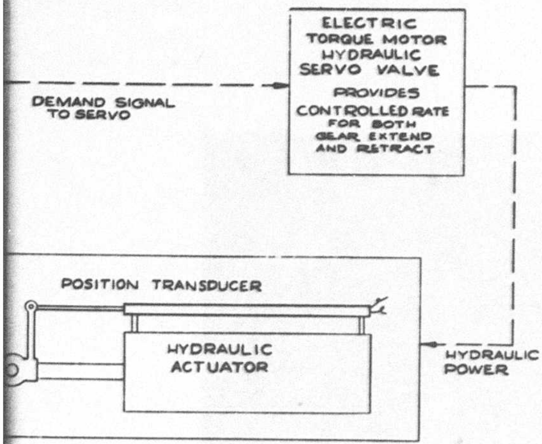
Figure 29. Mechanical Power Extraction System

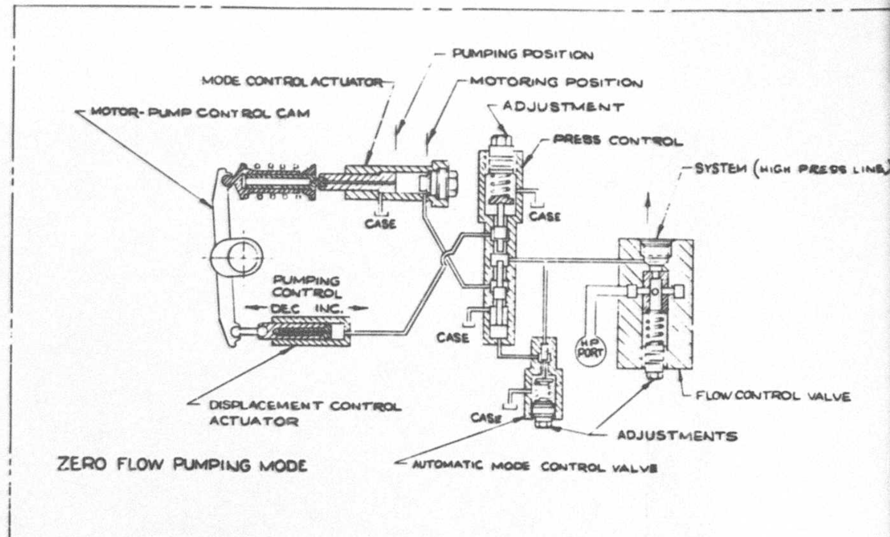


INTERMITTENT DUTY CYCLE SIMULATOR
MECHANICAL POWER EXTRACTION

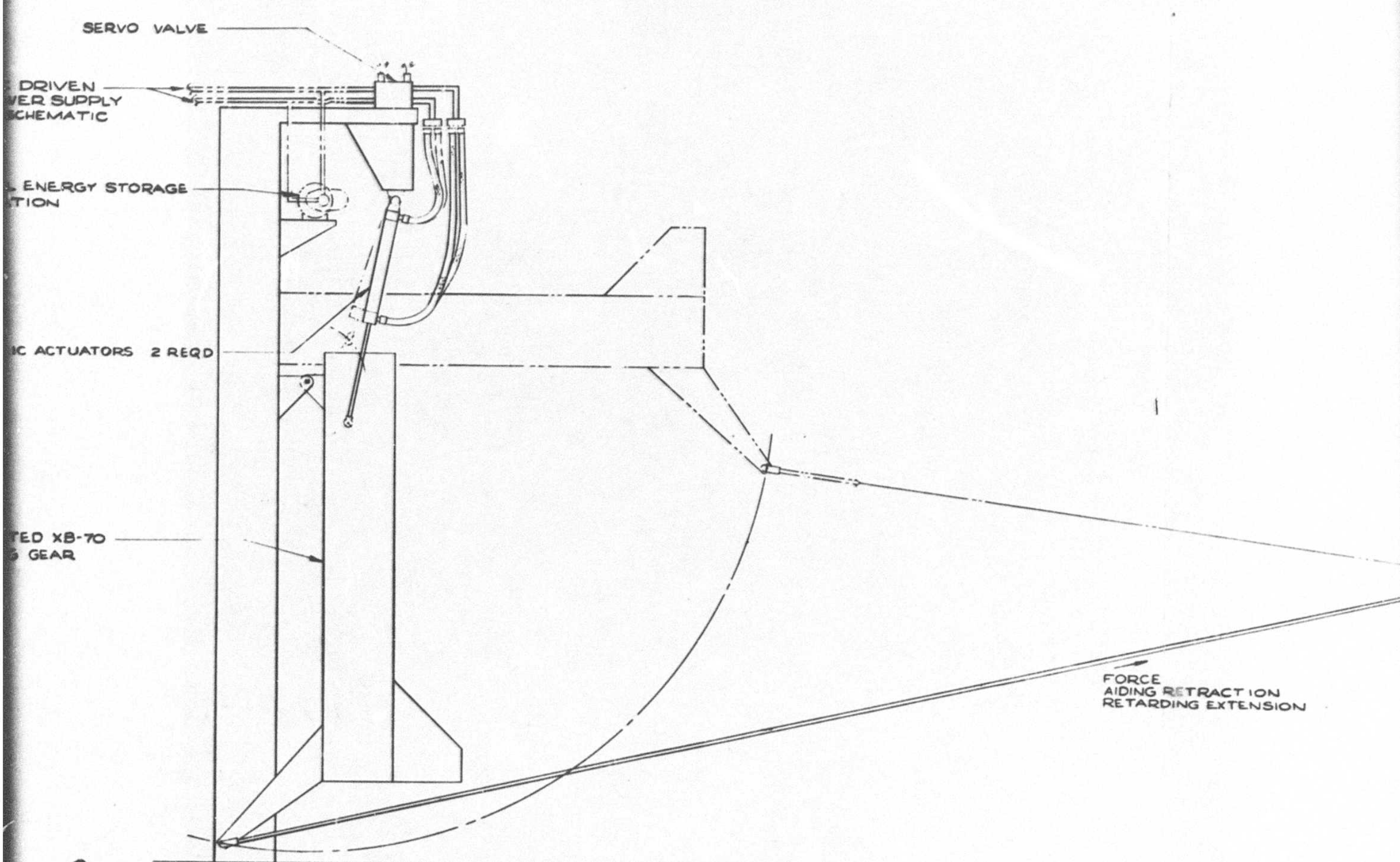
IN
H

A



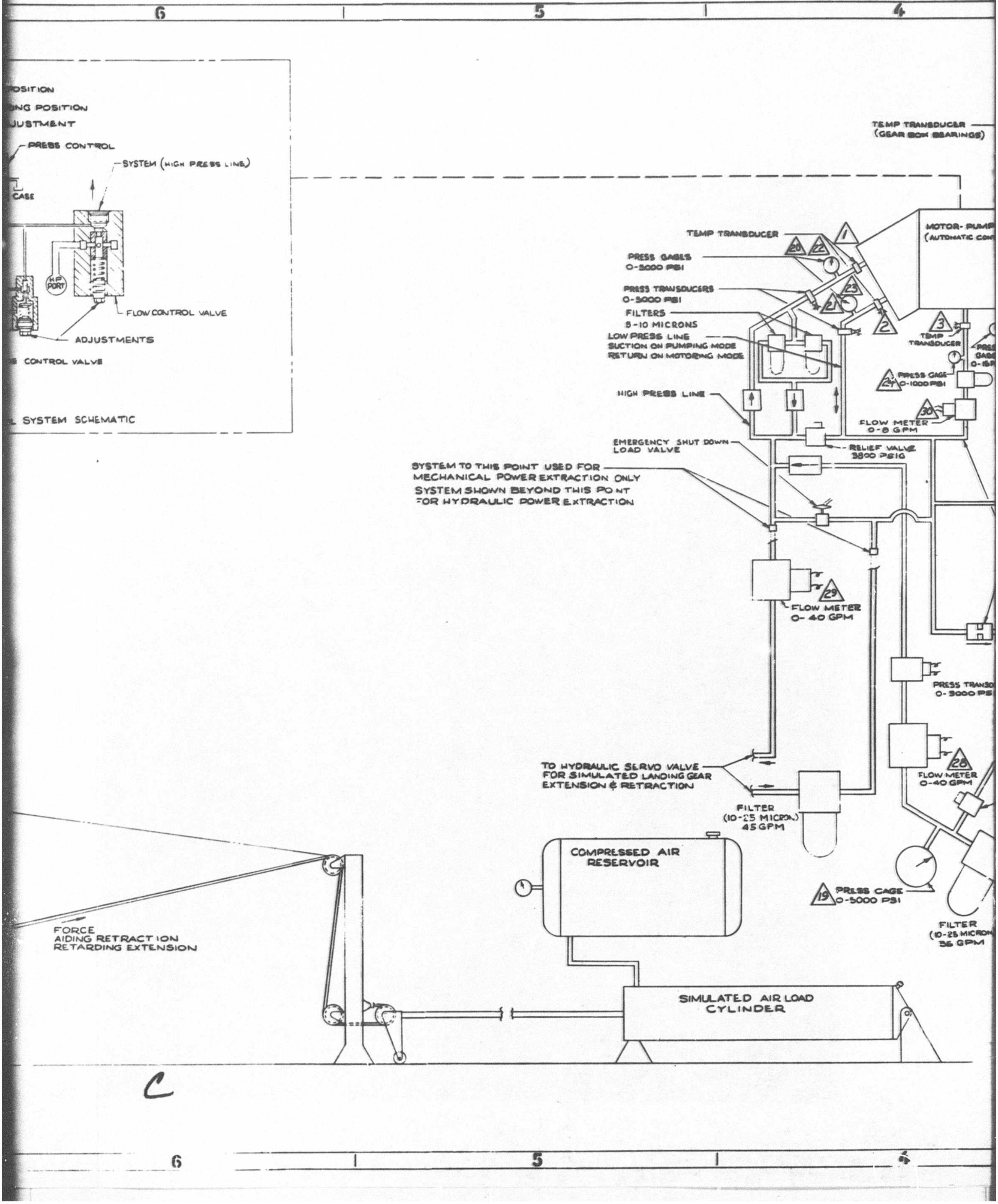


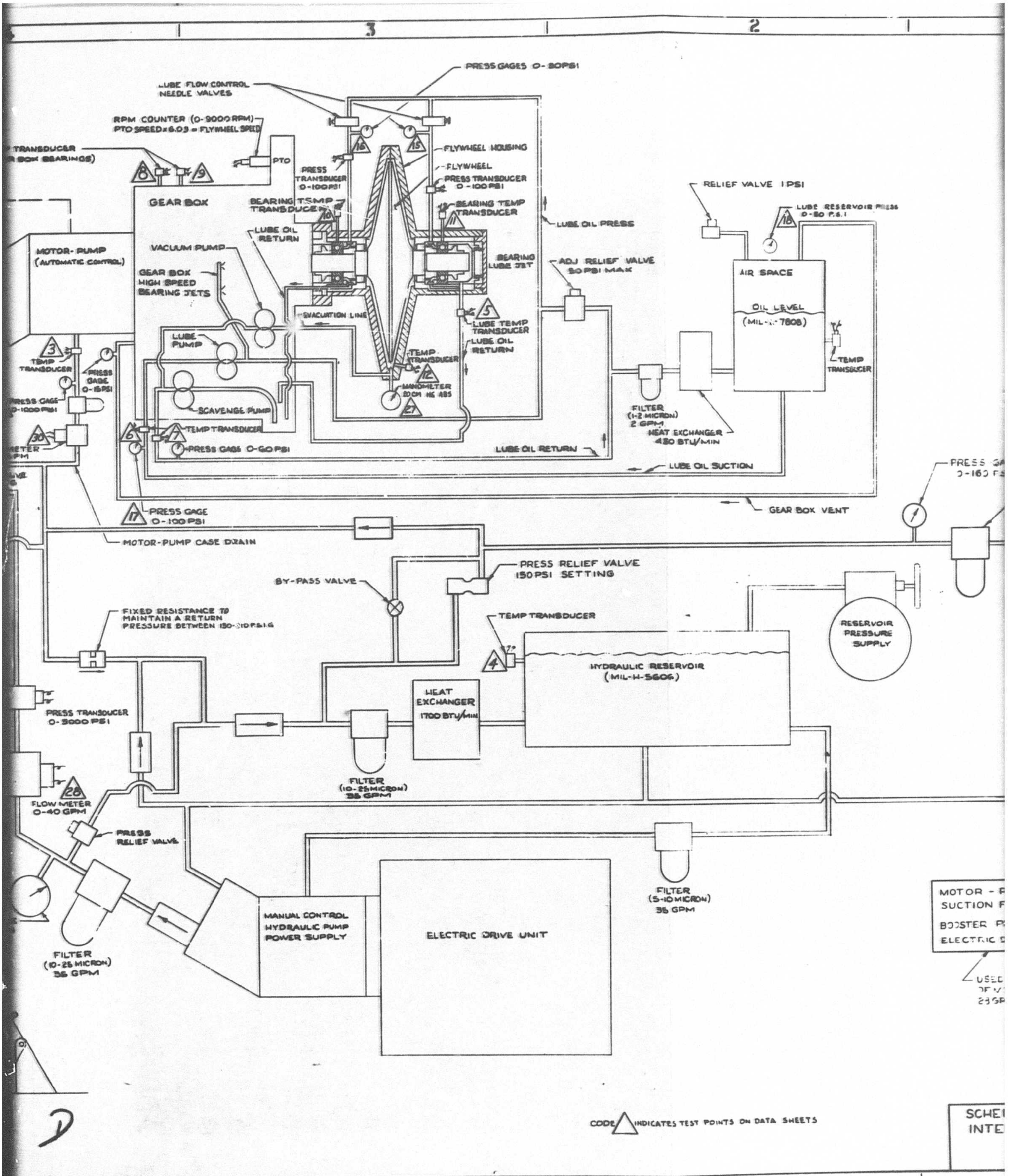
AUTOMATIC MOTOR-PUMP CONTROL SYSTEM SCHEMATIC



B

INTERMITTENT DUTY CYCLE SIMULATOR
HYDRAULIC POWER EXTRACTION


6
5
4
POSITION
NG POSITION
JUSTMENT
PRESS CONTROL
SYSTEM (HIGH PRESS LINE)
CASE
H.P. PORT
FLOW CONTROL VALVE
ADJUSTMENTS
CONTROL VALVE
SYSTEM SCHEMATIC
TEMP TRANSDUCER
(GEAR BOX BEARINGS)
MOTOR-PUMP
(AUTOMATIC CON
TEMP TRANSDUCER
PRESS GAGES
0-5000 PSI
PRESS TRANSDUCERS
0-5000 PSI
FILTERS
5-10 MICRONS
LOW PRESS LINE
SUCTION ON PUMPING MODE
RETURN ON MOTORING MODE
HIGH PRESS LINE
1
20
22
23
21
2
3
TEMP TRANSDUCER
PRESS GAGE
24
PRESS GAGE
0-1000 PSI
30
FLOW METER
0-8 GPM
EMERGENCY SHUT DOWN
LOAD VALVE
RELIEF VALVE
3800 PSIG
SYSTEM TO THIS POINT USED FOR
MECHANICAL POWER EXTRACTION ONLY
SYSTEM SHOWN BEYOND THIS POINT
FOR HYDRAULIC POWER EXTRACTION
29
FLOW METER
0-40 GPM
PRESS TRANSD
0-3000 PSI
28
FLOW METER
0-40 GPM
TO HYDRAULIC SERVO VALVE
FOR SIMULATED LANDING GEAR
EXTENSION & RETRACTION
FILTER
(10-25 MICRON)
45 GPM
COMPRESSED AIR
RESERVOIR
19
PRESS GAGE
0-5000 PSI
FILTER
(10-25 MICRON
35 GPM
FORCE
AIDING RETRACTION
RETARDING EXTENSION
SIMULATED AIR LOAD
CYLINDER
C
6
5
4


PRESS GAGES O- 80PSI
LUBE FLOW CONTROL NEEDLE VALVES
RPM COUNTER (0-9000 RPM)
PTO SPEED x 6.09 = FLYWHEEL SPEED
PTO
FLYWHEEL HOUSING
FLYWHEEL
PRESS TRANSDUCER 0-100 PSI
PRESS TRANSDUCER 0-100 PSI
GEAR BOX
BEARING TEMP TRANSDUCER
BEARING TEMP TRANSDUCER
LUBE OIL RETURN
LUBE OIL PRESS
MOTOR- PUMP (AUTOMATIC CONTROL)
VACUUM PUMP
GEAR BOX HIGH SPEED BEARING JETS
BEARING LUBE JET
ADJ RELIEF VALVE 50 PSI MAX
EVACUATION LINE
LUBE PUMP
LUBE TEMP TRANSDUCER
LUBE OIL RETURN
TEMP TRANSDUCER
MANOMETER 20 CM HG ABS
SCAVENGE PUMP
TEMP TRANSDUCER
PRESS GAGE O-60 PSI
LUBE OIL RETURN
LUBE OIL SUCTION
GEAR BOX VENT
RELIEF VALVE 1 PSI
LUBE RESERVOIR PRESS 0-50 P.S.I
AIR SPACE
OIL LEVEL
(MIL-L-7808)
TEMP TRANSDUCER
FILTER (1-2 MICRON) 2 GPM
HEAT EXCHANGER 430 BTU/MIN
TEMP TRANSDUCER
PRESS GAGE 0-15 PSI
PRESS GAGE 0-1000 PSI
PRESS GAGE 0-100 PSI
MOTOR-PUMP CASE DRAIN
PRESS RELIEF VALVE 150 PSI SETTING
BY-PASS VALVE
FIXED RESISTANCE TO MAINTAIN A RETURN PRESSURE BETWEEN 130-210 P.S.I.G
TEMP TRANSDUCER
HYDRAULIC RESERVOIR (MIL-H-5606)
RESERVOIR PRESSURE SUPPLY
PRESS TRANSDUCER 0-3000 PSI
HEAT EXCHANGER 1700 BTU/MIN
FILTER (10-25 MICRON) 36 GPM
FLOW METER 0-40 GPM
PRESS RELIEF VALVE
FILTER (5-10 MICRON) 35 GPM
MANUAL CONTROL HYDRAULIC PUMP POWER SUPPLY
ELECTRIC DRIVE UNIT
FILTER (10-25 MICRON) 36 GPM
CODE INDICATES TEST POINTS ON DATA SHEETS
D

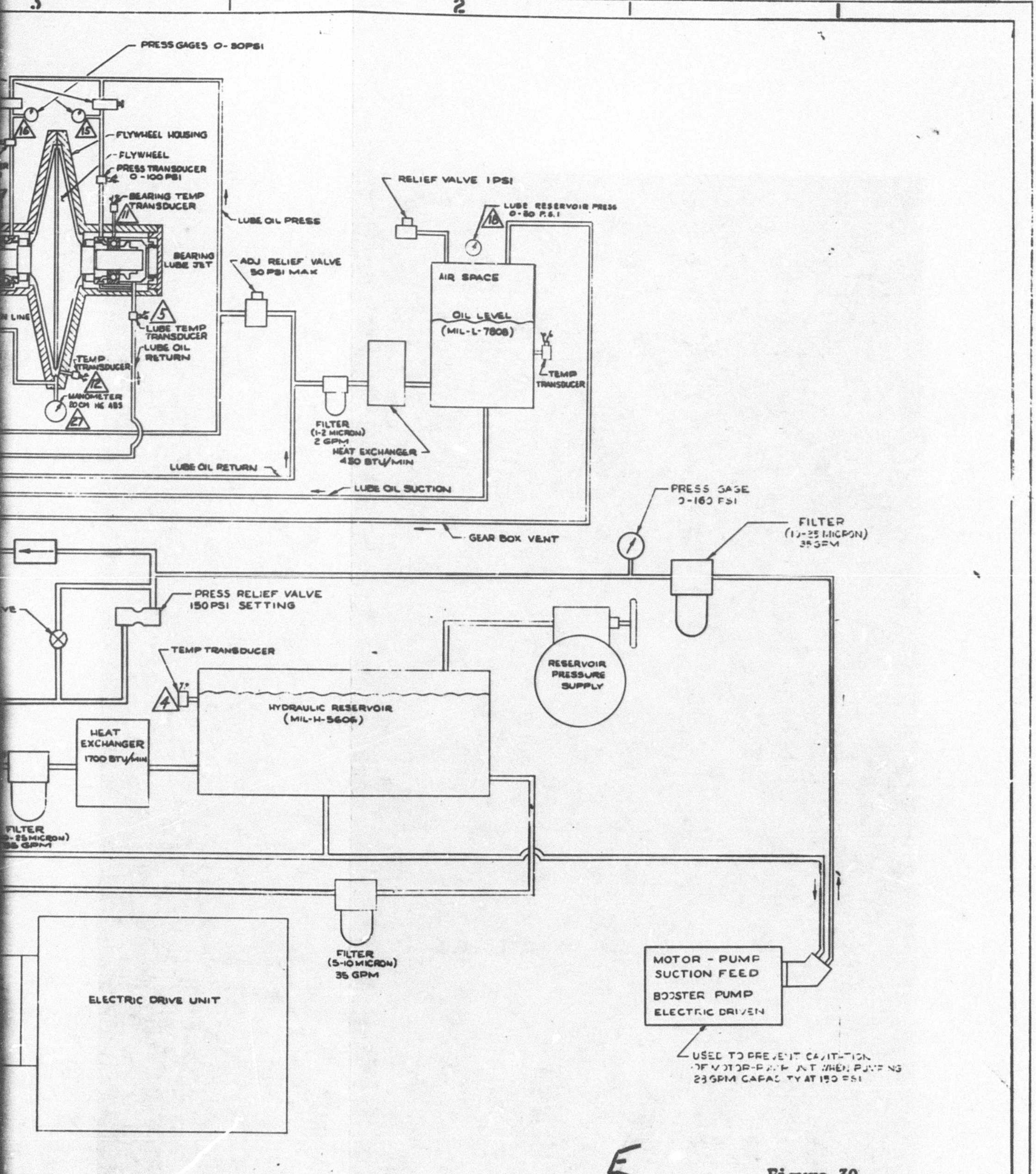


Figure 30.

SCHEMATIC - ENERGY STORAGE SUBSTATION
INTERMITTENT DUTY CYCLE SIMULATOR
HYDRAULIC AND MECHANICAL
POWER EXTRACTION

to one of the hydraulic actuators. A comparison of the two signals then established an error signal which operated the servo valve controlling the hydraulic supply to the strut actuators.

The principal advantage of using this system, in addition to accurate control, was in the safety provisions which resulted. If the landing gear did not follow the programed actuation cycle, the servo valve closed, stopping the landing gear. This allowed laboratory personnel to release the airload and lower the gear to the down position as a result of its weight.

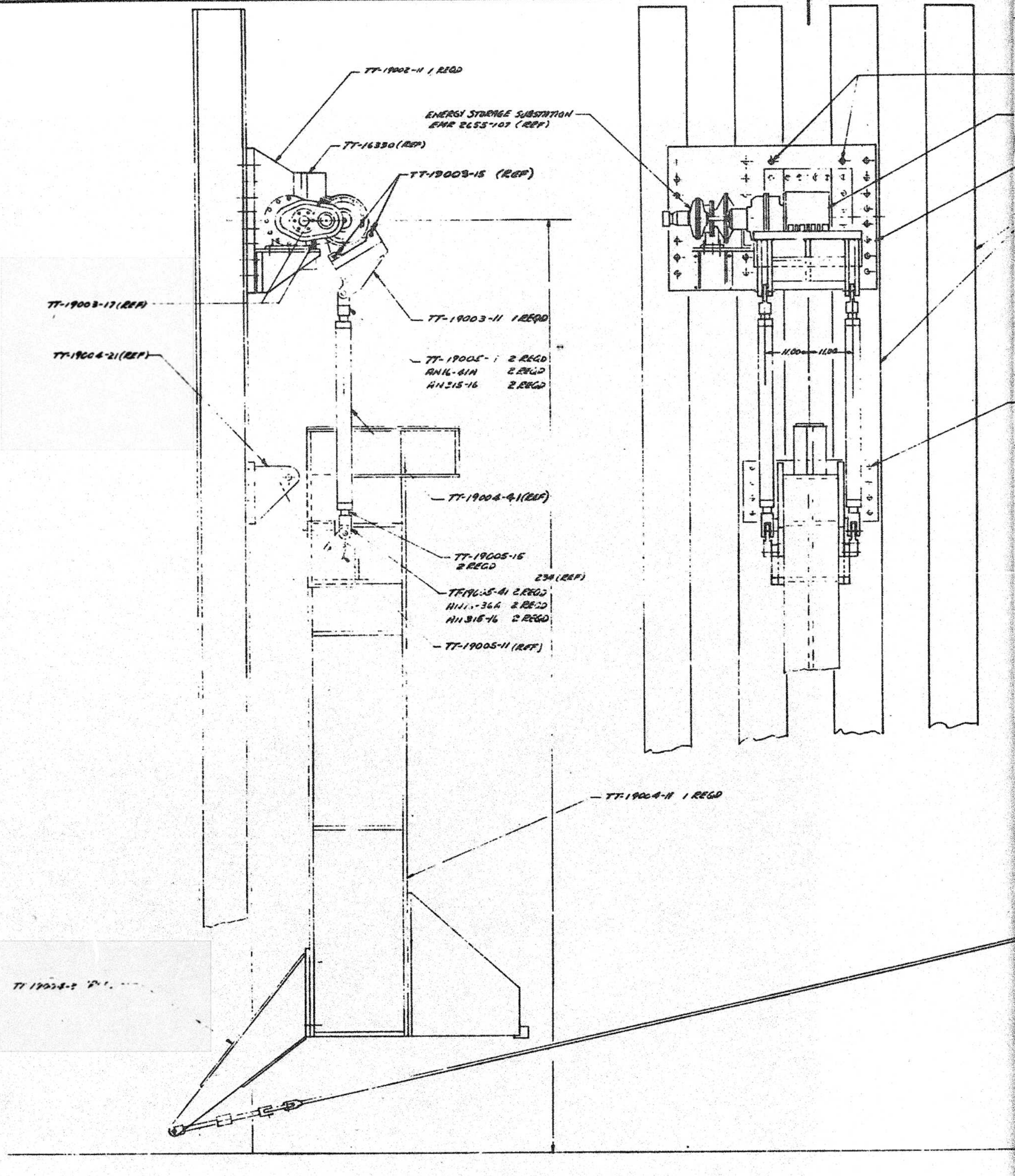
Figure 31 is an installation drawing showing both the hydraulic power extraction system and the mechanical power extraction system test setups. With reference to the drawing, the airload was produced by a pneumatic cylinder connected to a large accumulator or storage bottle, which maintained nearly constant pressure. The cable attachment was so located that it would vary the effective moment arm and produce the landing gear trunnion hinge moments shown by the dotted line in figure 32. This moment, when combined with that caused by the landing gear weight and fed through the kinematics closely approximated the requirements.

It will also be noted that the hydraulic power extraction system utilized two (3,000 psi) hydraulic actuators in tandem in lieu of the single (4,000 psi) hydraulic actuator used on the XB-70 air vehicle. When a single (3,000 psi) actuator was considered for this simulator it became apparent that a large side rotating moment would have to be resisted by the trunnion bearings since it (the 3,000 psi actuator) could not be installed on the centerline of the smaller XB-70 actuator. For the 10,000 cycles planned in the endurance test, the dual actuator arrangement appeared more practical.

The endurance test of 10,000 cycles was begun 18 May 1967 and completed 24 June 1967. In attaining this objective, Energy Storage Substation No. 3 operated 132 hours. It had only one failure, which involved the motor-pump. This was near the beginning of the test. The gearbox and flywheel completed the test without incident.

Figures 33 through 37 are photographs showing various views of the test setup. It should be noted that figures 35, 36, and 37 show the emergency generator motor. This unit was not used during the test. The pictures were taken prior to its destruction on the Tilt Table Simulator. The failure is described in a separate section covering operation of that simulator.

Operation of the simulator began using the first delivered automatic New York Airbrake motor-pump after it has been refurbished. This motor-pump failed for the second time after 12 hours (950 cycles) of operation and was replaced with the second automatic New York Airbrake unit. Shortly after this unit began operation it too began discharging abnormal quantities of chips through its case drain line. Although New York Airbrake representatives felt that the conditions under which this motor-pump, and most of its predecessors, had been operated was satisfactory and could not account for the failures, it was decided to increase the low case pressures (0 to 10 psi) by replumbing the simulator so that case pressure equaled return pressure (100 to 200 psi). As soon as this was done, the rate of chip generation decreased dramatically to normal limits and remained there for the rest of



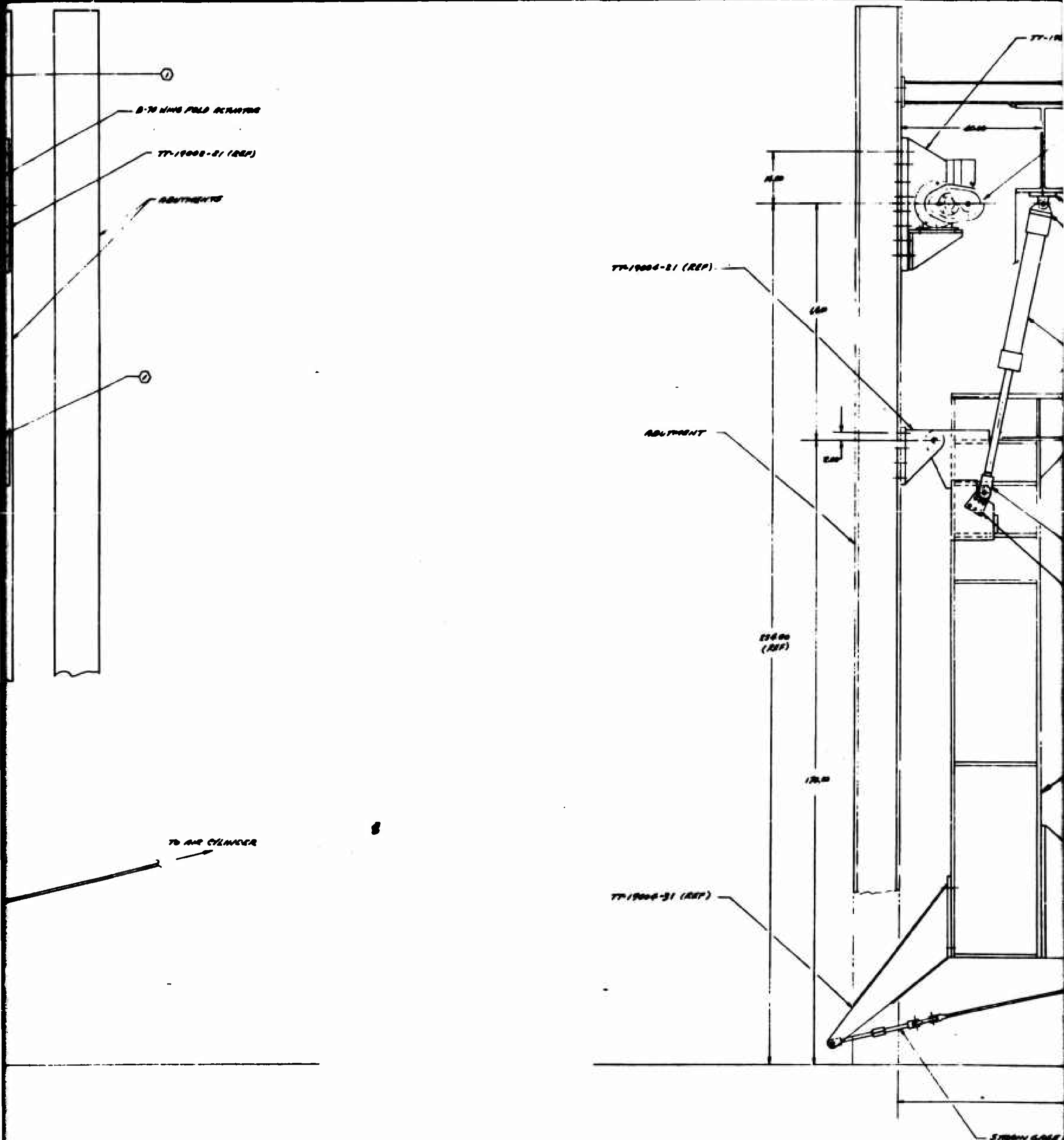
NORTH AMERICAN AVIATION, INC.
LOS ANGELES DIVISION

FRAME

TT-19001

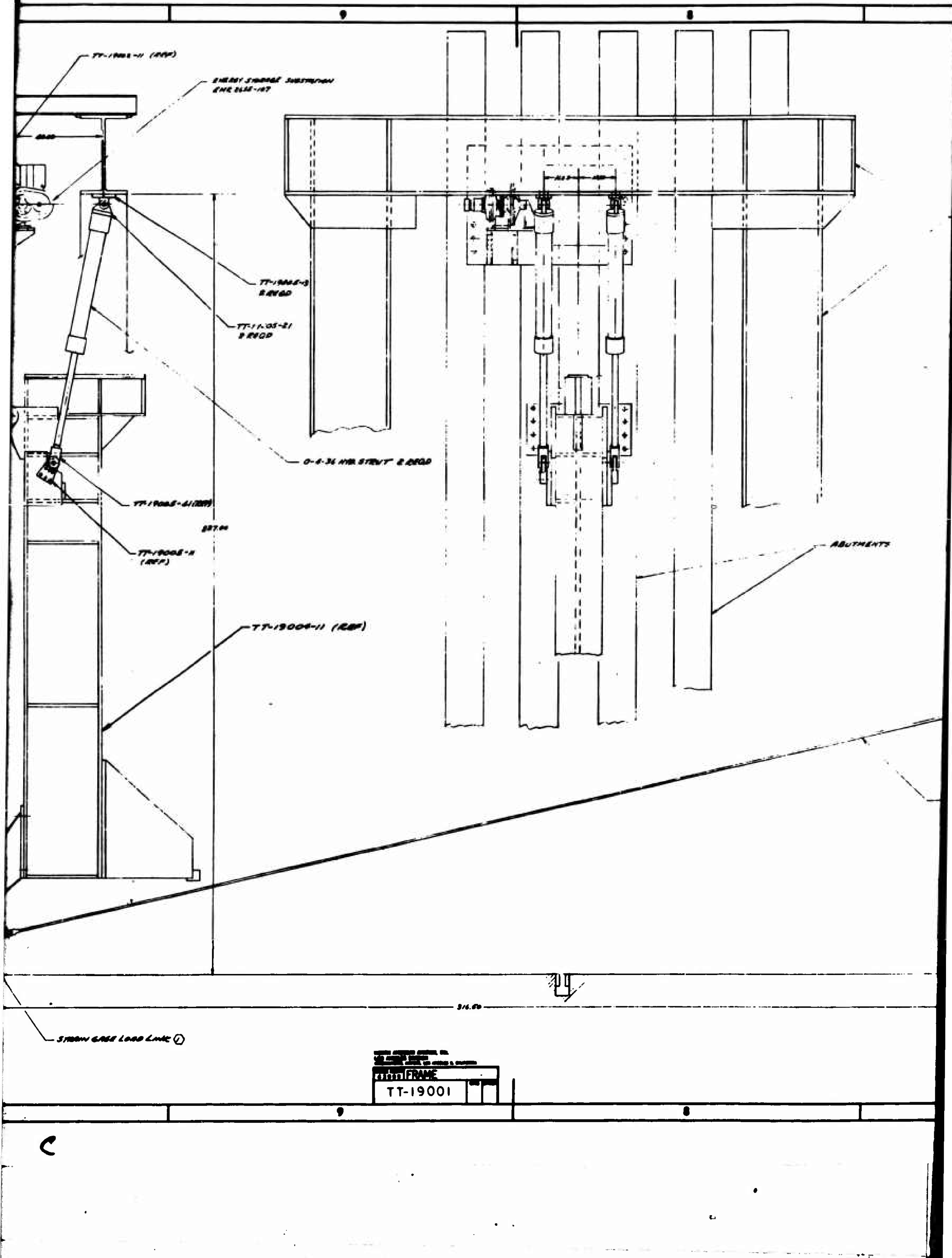
A

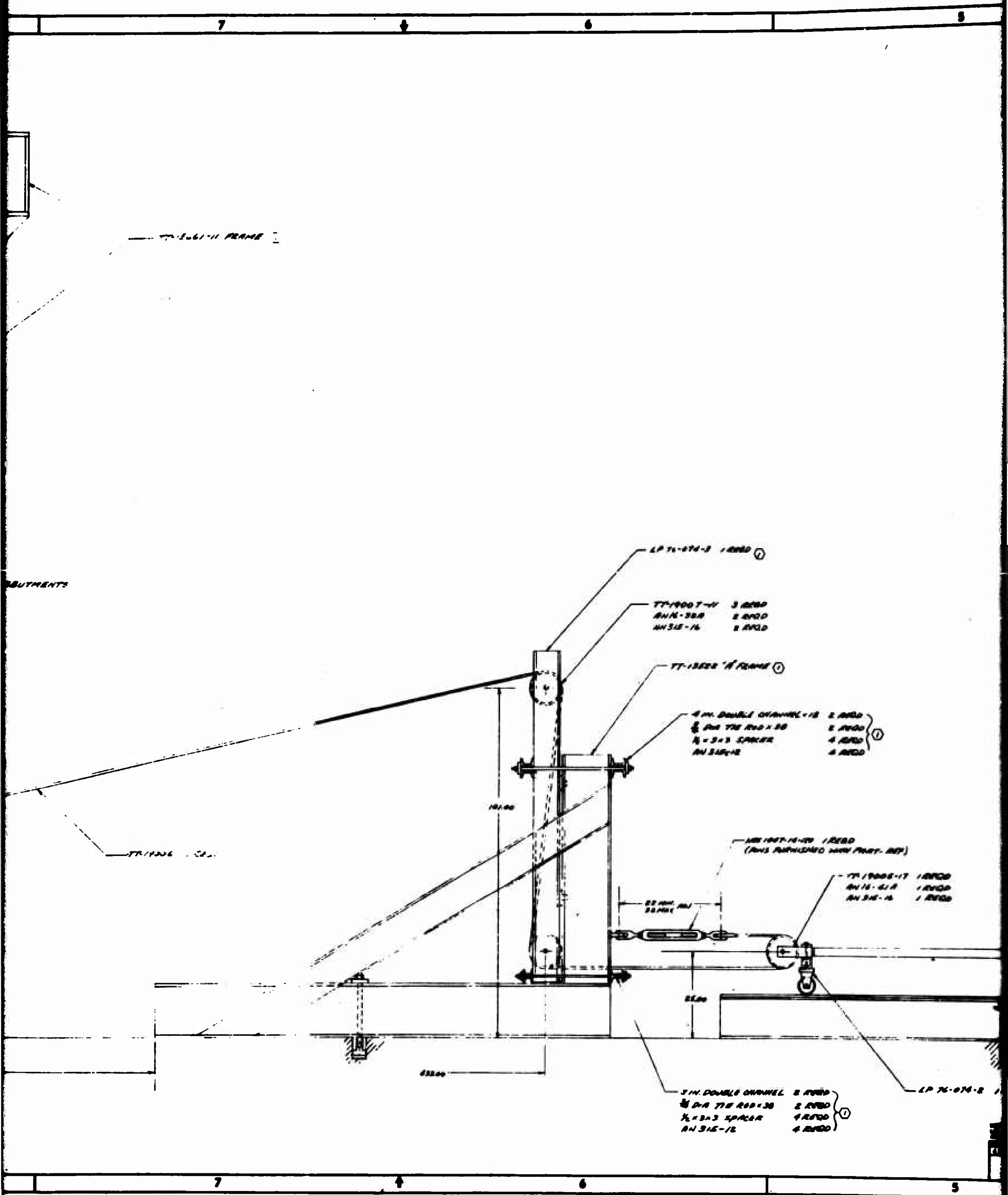
12 11 10



12 11 10

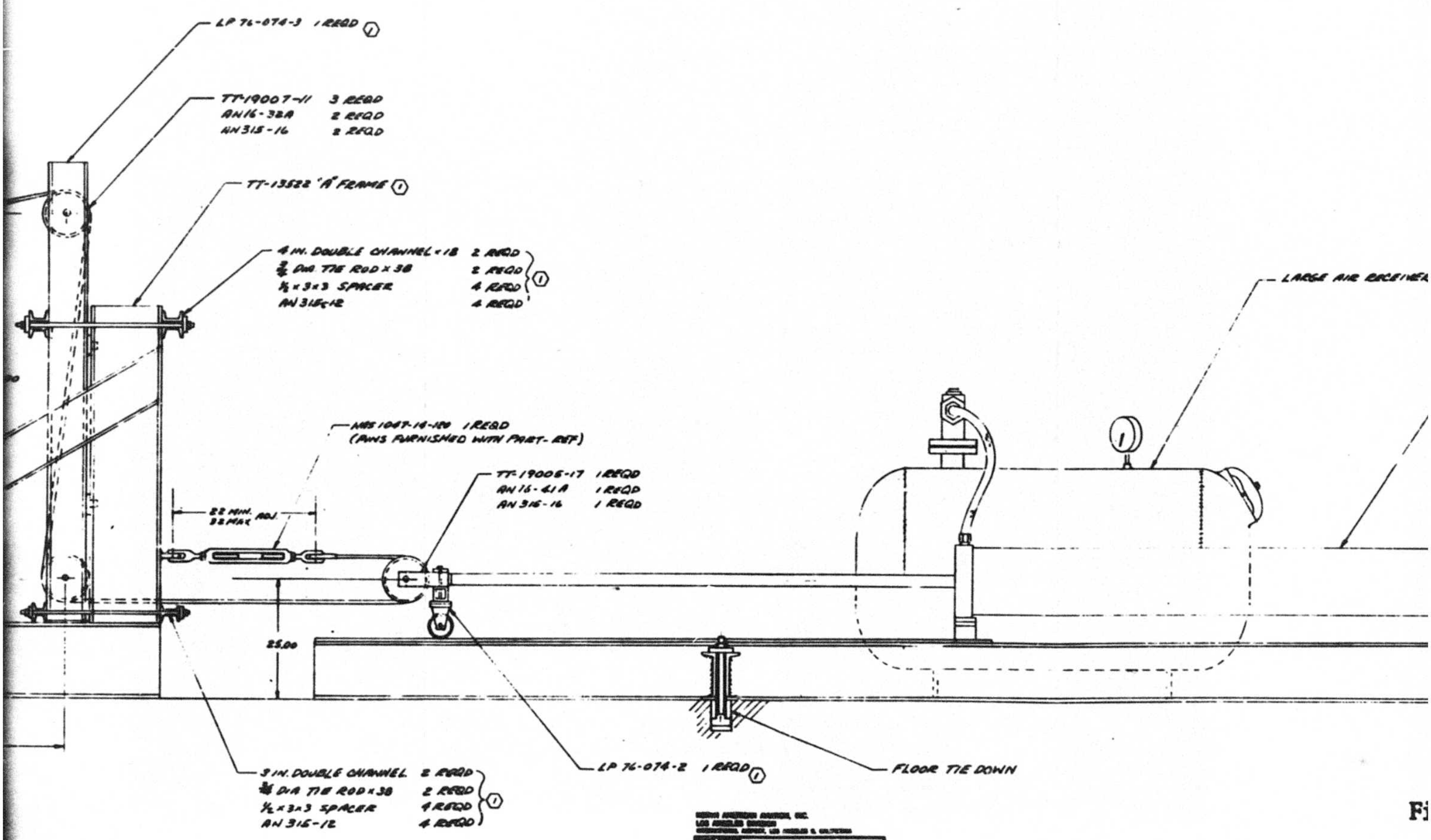
B


9
8
D-4-36 HYD. STRUT 2 REQD
TT-19004-11 (REF)
ABUTMENTS
227.00
316.60
STRAIN GAGE LOAD LINK (1)
FRAME
TT-19001
C



FRAME
ABUTMENTS
LP 76-074-3 1 REQD
AN 316-16 2 REQD
4 IN. DOUBLE CHANNEL×18 2 REQD
3×3 SPACER 4 REQD
AN 316-12 4 REQD
3 IN. DOUBLE CHANNEL 2 REQD
DIA TIE ROD×30 2 REQD
3×3 SPACER 4 REQD
AN 316-12 4 REQD
LP 76-074-2

6 5 4



LP 76-074-3 1 REQD 1
TT-19007-11 3 REQD
AN16-32A 2 REQD
AN315-16 2 REQD
TT-13522 "A" FRAME 1
4 IN. DOUBLE CHANNEL × 18 2 REQD
3/4 DIA TIE ROD × 38 2 REQD
1/2 × 3 × 3 SPACER 4 REQD
AN 316-12 4 REQD
1
LARGE AIR RECEIVER
MS 1047-14-120 1 REQD
(PINS FURNISHED WITH PART- REF)
TT-19006-17 1 REQD
AN 16-41 A 1 REQD
AN 316-16 1 REQD
22 MIN.
32 MAX
ADJ.
25.00
3 IN. DOUBLE CHANNEL 2 REQD
3/4 DIA TIE ROD × 38 2 REQD
1/2 × 3 × 3 SPACER 4 REQD
AN 316-12 4 REQD
1
LP 76-074-2 1 REQD 1
FLOOR TIE DOWN
FRAME
TT-19001

Fi

6 5 4

E

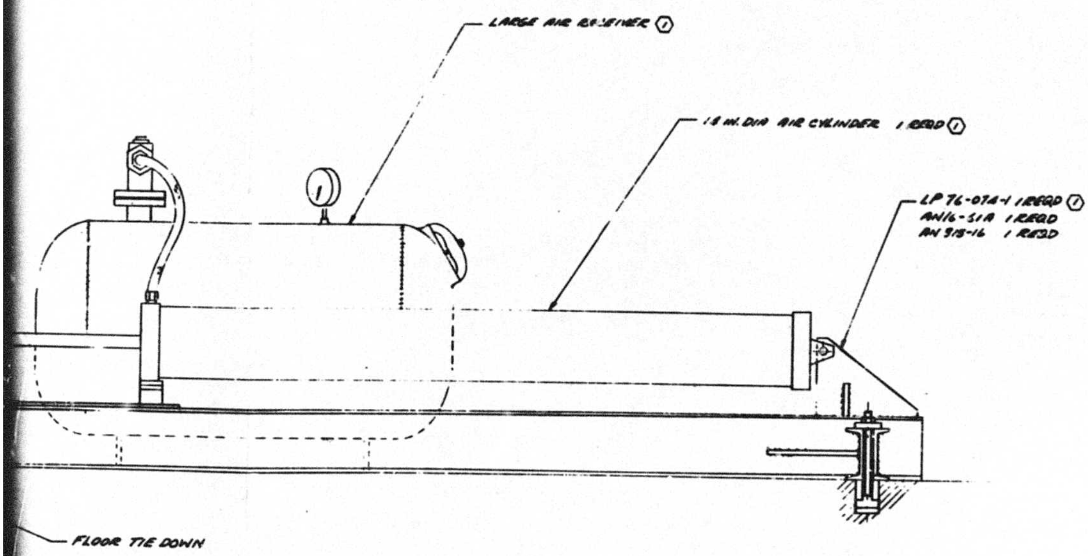


Figure 31. Installation - Energy Storage Substation Landing Gear Simulator

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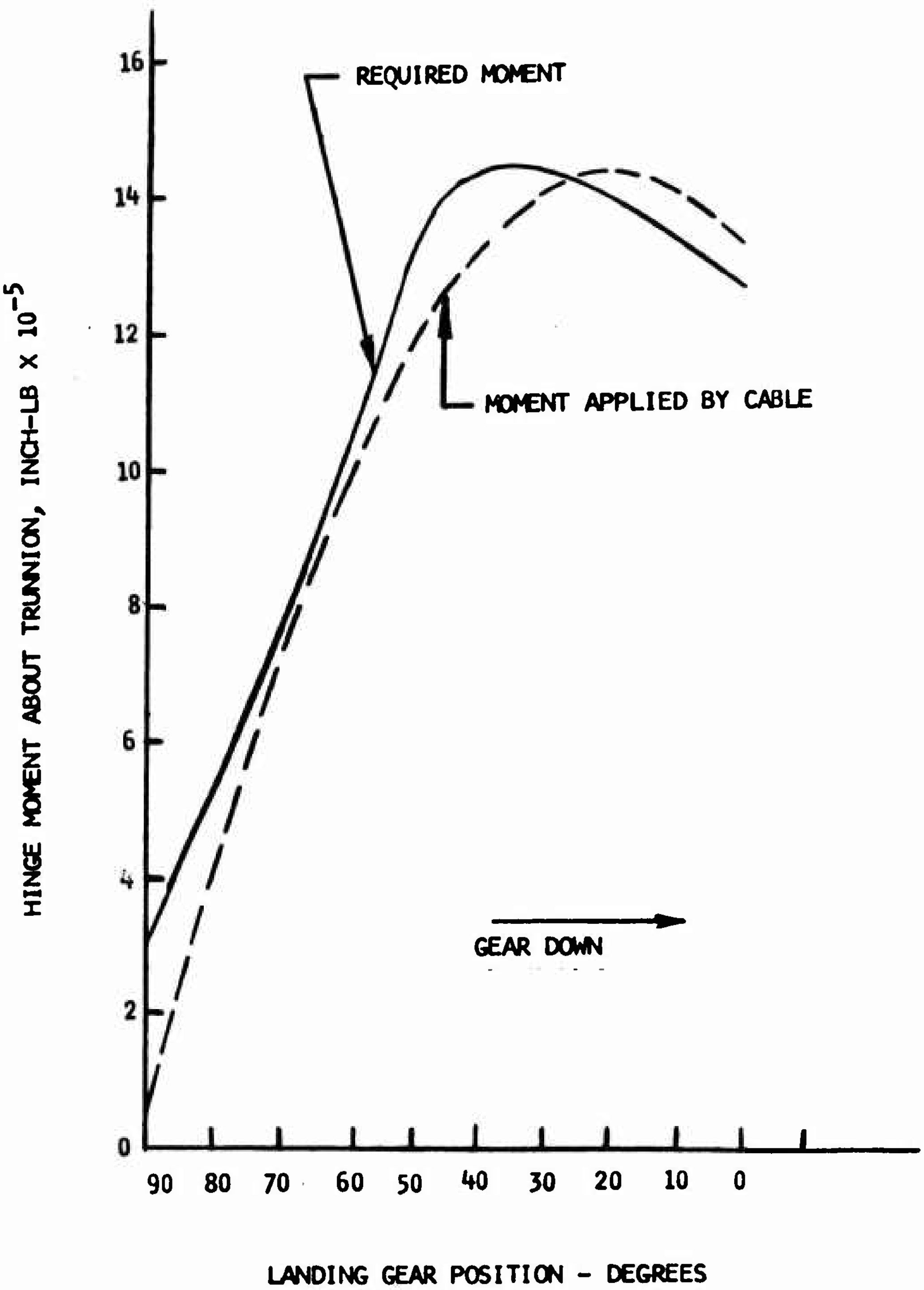


Figure 32. Simulated Airload on Landing Gear

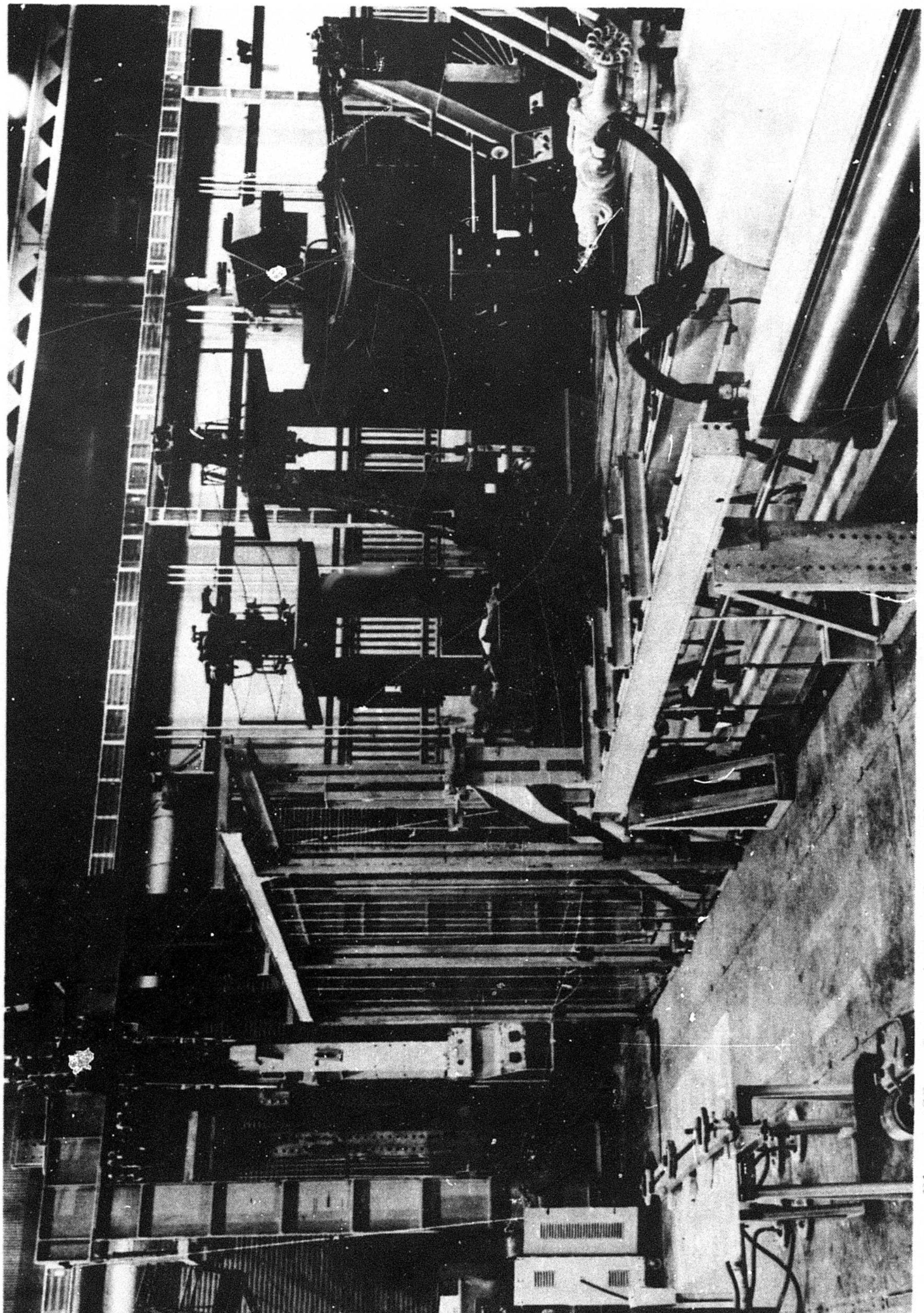


Figure 33. Intermittent Duty Simulator - Panoramic View Showing Air Load Cylinder

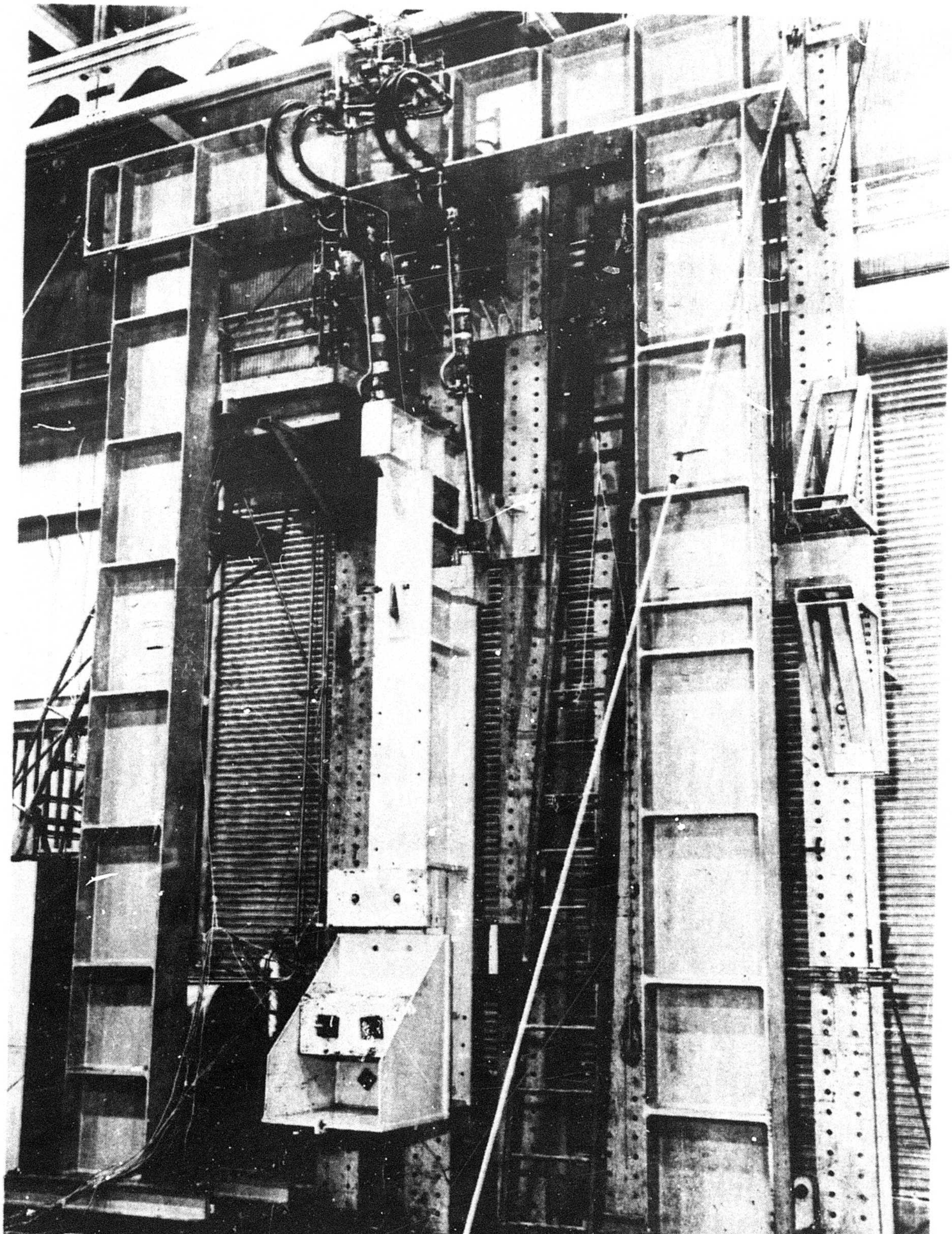


Figure 34. Intermittent Duty Simulator - Hydraulic Installation

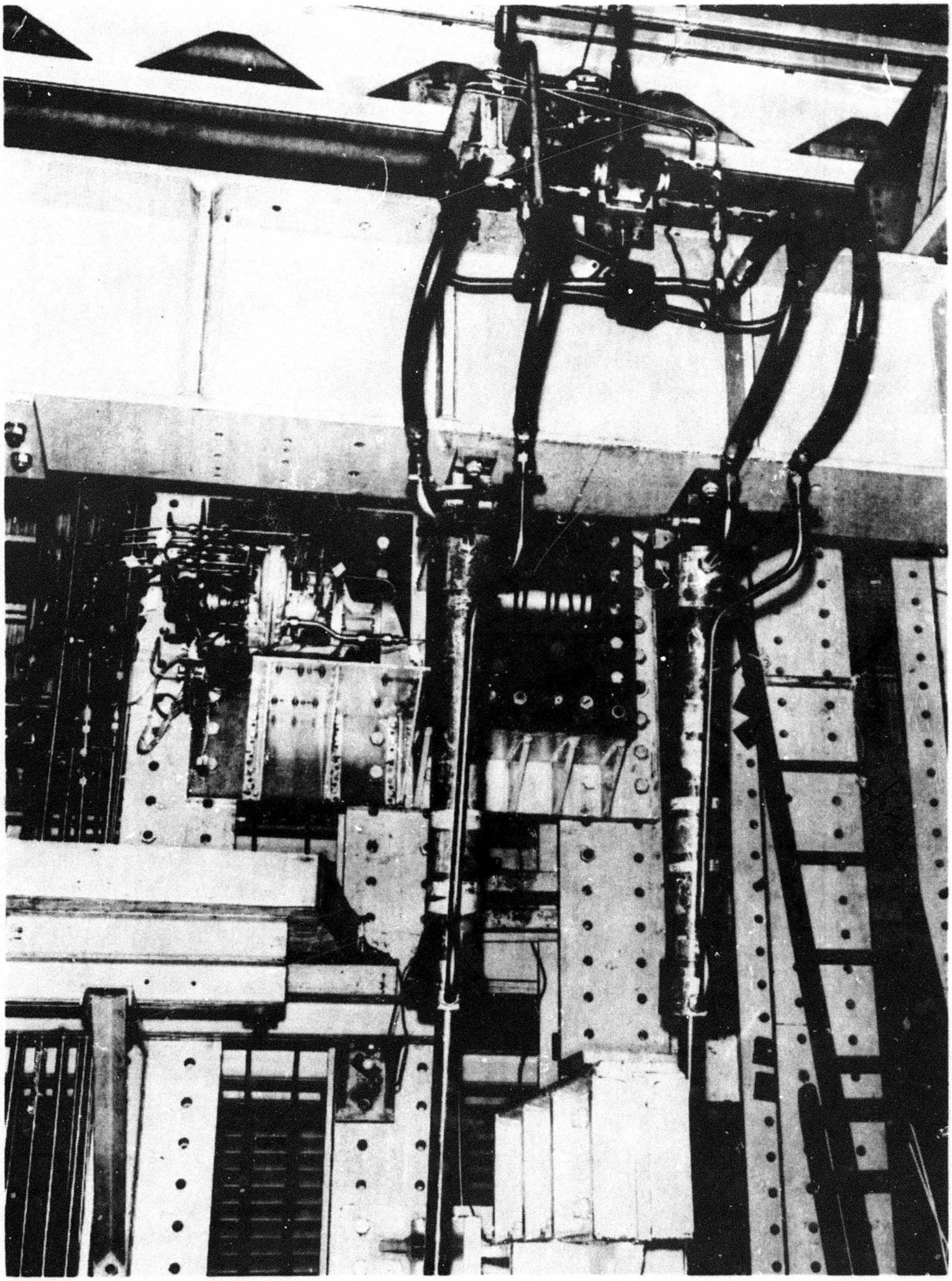


Figure 35. Intermittent Duty Simulator - Hydraulic Installation Closeup

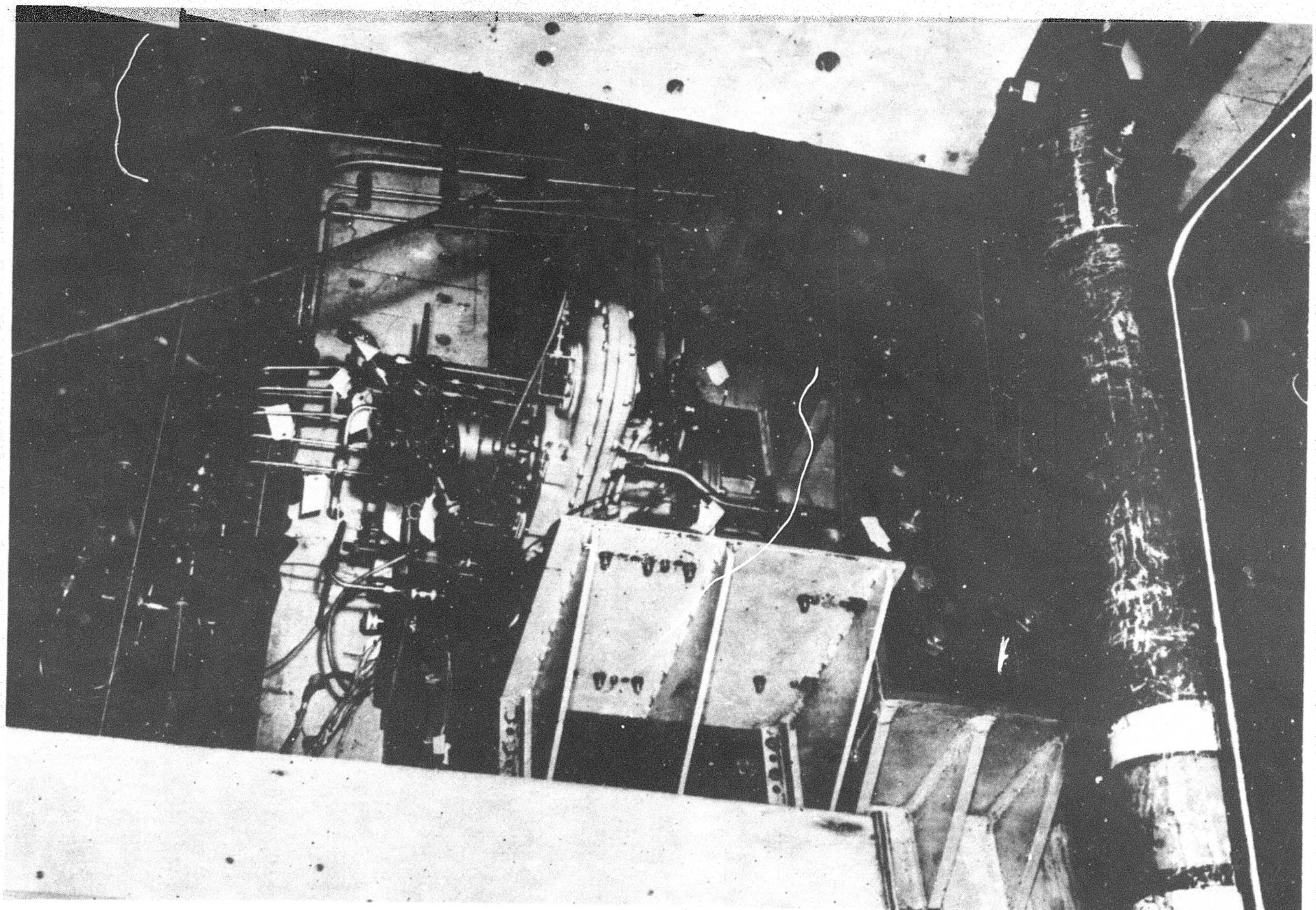


Figure 36. Intermittent Duty Simulator - Energy Storage Substation Close-up

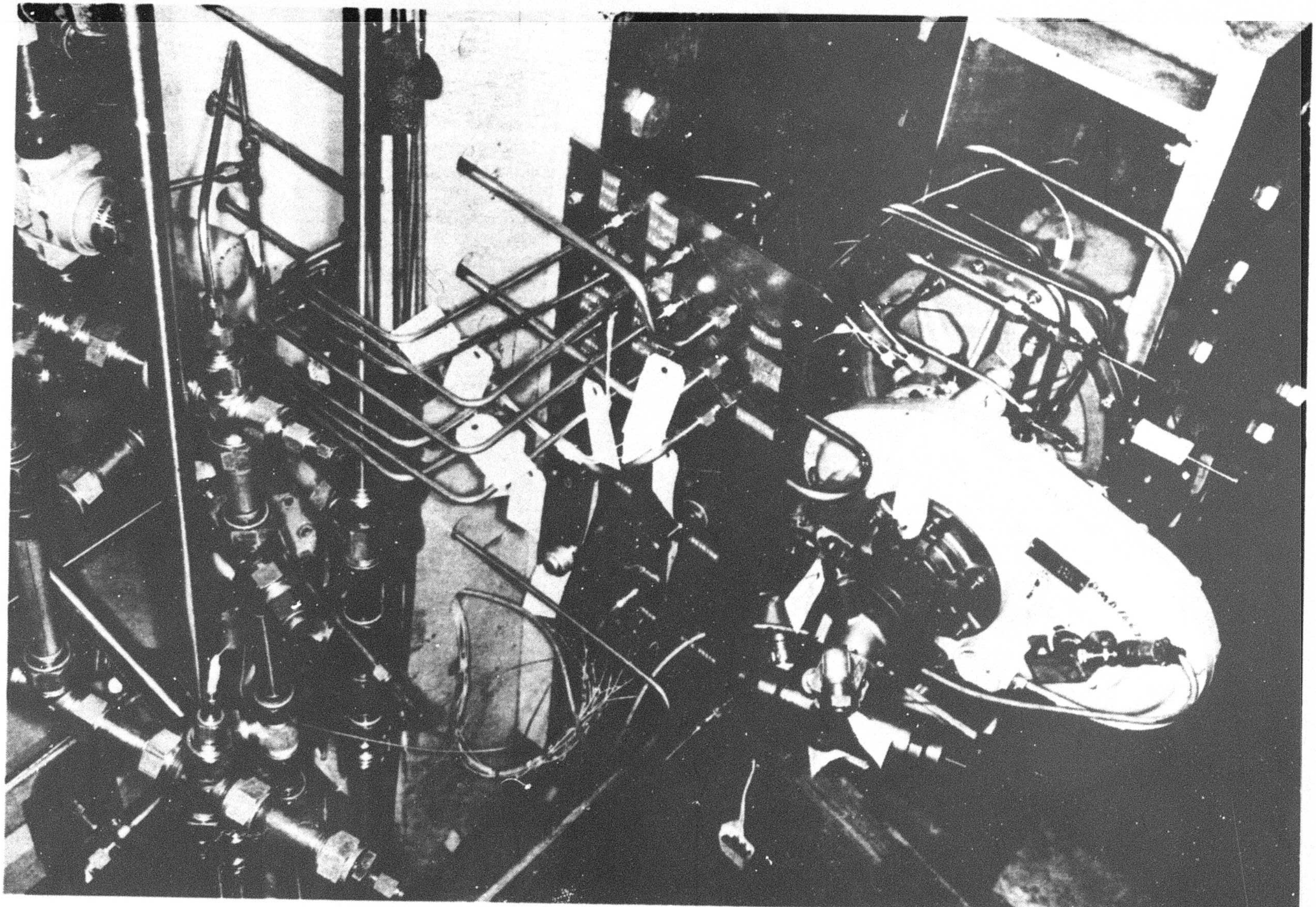


Figure 37. Intermittent Duty Simulator - Energy Storage Substation Closeup (Front View)

the tests. At present, there is still no satisfactory analytical explanation for the improvements resulting from this change. However, it should be pointed out that, from that time until the end of the program all motor-pump operation used the high case pressure approach and no further motor-pump failures occurred. In fact, from that time forward, motor-pump performance was excellent.

At the beginning of the endurance test the flywheel had three manometers. One read flywheel hub absolute pressure, one read flywheel differential from the hub to the tip and a third read baraometric pressure at the flywheel tip. During the initial stages of simulator operation it was observed that these manometer lines connected with the flywheel housing would gradually fill with oil. This led to false manometer readings. It was believed that this phenomenon stemmed from peculiarities of the vacuum system. Upon shutdown, the vacuum pump discharge line would siphon a small amount of return oil backward through the meshing gear teeth of the vacuum pump due to residual vacuum in the flywheel housing. Upon startup the bulk of this oil would gradually be scavenged from the housing by the flywheel rotation. In the process, however, the high tip speed of the wheel would vaporize some of the oil it contacted and send the vapors into all parts of the vacuum system. The manometer lines being lower than the substation by about 25 feet, and cooler due to room temperature exposure, would recondense the vapors, trapping the liquid on top of the columns of mercury. The condition was remedied by deleting two of the manometers which were considered nonessential and installing the third manometer above the substation. It supplied absolute pressure at the flywheel tip and was read through a telescope near the test panel. An electrically operated dump valve was also installed at the drain line from the flywheel housing. The dump valve was operated just as substation shutdown was initiated. The line was left open until the next startup and only a small amount of dripping occurred. This was believed to be exclusively rotary seal leakage, since a slight amount of carbon discoloration could usually be detected. With operation as outlined above a close surveillance of the condition of the rotary seals was possible throughout the endurance test. The surveillance indicated that there was no significant change in leakage throughout the test.

During the course of testing most of the interruptions resulted from failures in the landing gear test fixture and its associated commercial and/or instrumentation equipment. The commercial hydraulic actuators, which raise and lower the simulated landing gear strut, were initially designed with aluminum pistons that seal to the barrel by means of an O-ring that is banded on its outside diameter by a teflon ring or "slipper". The nature of the seal installation coupled with the manner in which the actuator was used allowed extrusion of the seal through the piston-to-barrel clearance. An excessive amount of clearance had built up as a result of the actuator's previous usage. This fact plus the large moment on the rod end tending to cock the piston in the barrel led to piston seal extrusion. The same large moment conditions also led to several actuator rod seal failures. Prior to beginning the endurance test, during simulator fabrication, the actuators were refurbished with new seals. New piston and/or rod seals were also installed after 2,756, 4,030, 5,734, and 8,505 cycles due to leakage or excessive bypassing of fluid past the pistons. The condition was improved during the

course of the refurbishments by machining new pistons which had less clearance, and honing the barrels to a very smooth finish. The rod end and cylinder head pivot bearings on these actuators began to gall and bind between the bolt and bearing inside diameter after only a few cycles. Bearing and bolt replacements were made but this only brought temporary relief. The bolts were then drilled and fitted for grease injection. This change, together with periodic grease injection (every 100 cycles), corrected the problem.

After approximately 5,200 cycles, it was observed that a strand of the steel cable which applies the equivalent airload to the strut had broken. This signaled the beginning of cable deterioration, and the cable assembly was therefore replaced. Since this is a relatively inexpensive item and since its failure could result in extensive damage to surrounding equipment, several spare cables were procured with the intent of periodic replacement throughout the remaining portions of the endurance tests. (This cable arrangement would be operative throughout the mechanical power extraction endurance tests as well as those of the hydraulic power extraction.)

Prior to beginning Intermittent Duty Cycle Simulator testing, it became apparent that it would be necessary to operate at landing gear speeds somewhat above that which are used by the XB-70 air vehicle. (The XB-70 air vehicle requires 12.5 seconds to extend the strut and 7.5 seconds to retract the strut.) The Intermittent Duty Cycle Simulator was operated with a 6-second extend time and a 5-second retract time. The dwell time in the up position was 16-1/2 seconds while the dwell time in the extended position was 13 seconds.

This speedup in rate resulted from the fact that the displacement of the motor-pump unit as a motor equals its displacement as a pump and the additional fact that the motor is a fixed (maximum) displacement device at all times when it is in a motoring mode. As a result of this fact the ground power unit (GPU) (the test equivalent of the engine-driven pumps) was required to supply sufficient flow to the motor to wind the flywheel up to 53,000 rpm (i.e., 13.5 gpm plus leakage). This flow, supplied to the two commercial cylinders used to raise and lower the landing gear, was sufficient, all by itself, to raise and lower the gear in the 12- to 15-second operating time range. Under these conditions no power extraction demands would be made on the energy storage substation. In order to draw power from the substation it would, therefore, either be necessary to increase the size of the actuating cylinders or reduce the operating time to approximately one half. It was elected to reduce the operating time to the aforementioned 6 seconds extend and 5 seconds retract.

The data from this test are shown in table XII (three pages). Figure 38 is a sample data record taken during a gear-down and gear-up cycle.

Based on the data gathered during the Intermittent Duty Cycle Endurance Test, table XIII was created to summarize performance characteristics. This summary indicates that the efficiency of extracting power during gear-down and gear-up is 58.7 and 44.5 percent, respectively. (Items t and u.) The losses which go into making up this efficiency can be categorized into those

Table XII

INTERMITTENT DUTY CYCLE SIMULATOR TEST DATA
(HYDRAULIC POWER EXTRACTION)

Data Sheet No. 1

No.	Item									
	Date		5-18	5-23	5-24	5-24	6-2	6-3	6-5	6-7
	Accum. Running Time of Test - Hours		1.0	3.2	7.9	13.0	19.4	25.1	30.9	37.4
	Total Operation Load Cycles		50	100	500	950	1500	2000	2500	3000
	Vibration Amp. at Flywheel Bearings g's	Inboard	7.3	6.7	6.7	6.5	5.3	5.9	5.7	4.9
		Outboard	7.3	7.9	7.8	7.8	7.4	7.4	7.7	6.4
	Flywheel Speed at Full Motoring - RPM		50400	52550	52550	51600	50800	50750	50950	51750
	Minimum Speed Attain Under Load - RPM		45018	45350	45950	45650	44650	44250	44750	45550
	Average Operational Time Sec./Cycle	Gear Up	8	5	5	5	5	5	5	5
		Gear Down	10	6	6	6	6	6	6	6
	Dwell Time - Seconds	Gear Up	25	13	13	13	13	13	13	13
		Gear Down	25	16-1/2	16-1/2	16-1/2	16-1/2	16-1/2	16-1/2	16-1/2
	Max. Load on Gear - lbs.	Gear Up	5519	5580	5720	5630	5775	5725	5770	5860
		Gear Down	7582	7635	7580	7590	7780	7635	7590	8010
NO.	TEMPERATURE DATA - DEG. F									
1	Test Motor - Pump Inlet Temp.		94	108	113	116	105	105	107	101
2	Test Motor - Pump Return Temp.		97	109	112	114	102	105	105	109
3	Test Motor - Pump Case Drain Temp.		133	147	146	144	145	148	146	147
4	Main Hyd. Reservoir Supply Temp.		99	124	118	118	99	103	102	101
5	Gear Box Lube Oil Return Temp.		170	185	180	177	192	180	169	194
6	Gear Box Lube Oil Suction Temp.		72	75	75	74	68	70	69	72
7	Gear Box Scavanger Oil Return Temp.		149	161	153	151	156	158	150	163
8	Gear Box Brg. Adj. Pump Temp.		158	173	167	165	170	171	175	173
9	Gear Box Brg. Adj. Flywheel Temp.		177	197	192	189	192	195	190	195
10	Inboard Flywheel Bearing Temp.		165	185	178	176	179	184	176	185
11	Outboard Flywheel Bearing Temp.		158	178	170	165	173	176	168	177
12	Flywheel Case Housing Temp.		152	172	154	154	172	171	150	175
NO.	SYSTEM PRESSURE DATA - PSIG									
13	Actuator Press - Tension-Gear Up		3048	3388	3120	3100	3115	3240	3020	3270
14	Actuator Press - Comp.-Gear Down		3093	3137	3060	3100	3340	3090	3050	3390
15	Outboard Flywheel Bearing Press.		38.0	39.8	39.2	39.7	40.7	41.1	41.6	40.1
16	Inboard Flywheel Bearing Press.		37.4	39.0	39.5	39.7	40.7	39.7	40.4	39.5
17	Lube Pump Pressure Port Press.		50	50	50	50	50	50	50	50
18	Lube Reservoir Press.		0	0	0	0	0	0	0	0
19	Hyd. Supply Pump Press. at Source		3400	3415	3390	3390	3340	3352	3450	3400
20	Test Motor - Pump Port Pressures (Pumping)	Press.	3080	3009	2550	24440	2910	2840	2900-2810	2920-2810
21		Return	66	129	157	168	173	172	175-222	156-206
22	Test Motor - Pump Port Pressures (Motoring)	Press.	3344	3539	3480	3520	3328	3310	3311	3330
23		Return	130	139	188	171	188	192	192	170
24	Test Motor - Pump Case Drain Press.		155-210	155-210	155-210	155-210	155-210	155-210	155-208	153-205
25	Vacuum at Flywheel Hub - in. Hg		29.6	—	—	—	—	—	—	—
26	Vacuum from Hub to Tip (ΔP) in. Hg		.6	—	—	—	—	—	—	—
27	Press. at Flywheel Hub - Cm. Hg. Ab.		1.5	3.2	3.6	3.5	3.3	3.5	3.5	3.5
NO.	FLUID FLOW DATA									
28	Hyd. Supply Pump Flow - GPM		17.6	18.3	18.0	19.3	18.7	19.0	20.2	18.5
29	Inlet Flow to Servo - GPM		20.8	32.0	31.2	31.4	31.3	34.0	35.2	33.6
30	Case Drain of Motor - GPM		.7	.65	.7	.65-.85	.3-.55	.3-.55	.3-.55	.3-.55
31	Outb'd. Bearing Lube Flow cc/min		550	550	555	575	595	595	595	595
32	Inb'd. Bearing Lube Flow cc/min		540	570	580	590	610	610	610	610
33	Axial Vib.		11.3	10.6	10.6	10.0	13.9	15.8	15.3	15.2

Table XII (Continued)

INTERMITTENT DUTY CYCLE SIMULATOR TEST DATA
(HYDRAULIC POWER EXTRACTION)

Data Sheet No. 2

No.	Item									
	Date		6-7	6-8	6-9	6-15	6-15	6-20	6-21	6-21
	Accum. Running Time of Test - Hours		43.3	49.5	55.7	62.1	68.6	74.9	80.8	86.9
	Total Operation Load Cycles		3500	4000	4500	5000	5500	6000	6500	7000
	Vibration Amp. at Flywheel Bearings g's	Inboard	5.3	5.3	6.7	6.3	5.5	5.9	12.6	6.0
		Outboard	6.7	8.8	7.8	7.4	6.8	7.2	12.9	7.1
	Flywheel Speed at Full Motoring - RPM		52650	51850	52050	53000	50800	51100	52050	52050
	Minimum Speed Attain Under Load - RPM		45550	45550	45450	46000	45400	45400	46500	46500
	Average Operational Time Sec./Cycle	Gear Up	5	5	5	5	5	5	5	5
		Gear Down	6	6	6	6	6	6	6	6
	Dwell Time - Seconds	Gear Up	13	13	13	13	13	13	13	13
		Gear Down	16-1/2	16-1/2	16-1/2	16-1/2	16-1/2	16-1/2	16-1/2	16-1/2
	Max. Load on Gear - lbs.	Gear Up	5710	6010	5440	5945	5350	5205	5450	5150
		Gear Down	7580	8010	7240	7920	7225	6940	7370	6980
NO.	TEMPERATURE DATA - DEG. F									
1	Test Motor - Pump Inlet Temp.		106	98	105	110	110	110	114	105
2	Test Motor - Pump Return Temp.		104	100	105	120	109	115	112	108
3	Test Motor - Pump Case Drain Temp.		145	142	150	147	146	147	150	150
4	Main Hyd. Reservoir Supply Temp.		100	98	105	105	105	110	106	105
5	Gear Box Lube Oil Return Temp.		190	165	161	187	178	165	185	205
6	Gear Box Lube Oil Suction Temp.		70	70	73	71	70	71	74	75
7	Gear Box Scavanger Oil Return Temp.		153	150	155	158	160	155	154	165
8	Gear Box Brg. Adj. Pump Temp.		167	163	167	170	173	165	169	185
9	Gear Box Brg. Adj. Flywheel Temp.		189	185	192	191	195	190	191	201
10	Inboard Flywheel Bearing Temp.		175	170	178	177	184	178	178	194
11	Outboard Flywheel Bearing Temp.		166	162	166	169	175	172	170	185
12	Flywheel Case Housing Temp.		150	140	144	148	170	147	154	180
NO.	SYSTEM PRESSURE DATA - PSIG									
13	Actuator Press - Tension-Gear Up		3280	3280	3300	3320	3295	3290	3260	3250
14	Actuator Press - Comp.-Gear Down		3260	3360	3340	3460	3340	3325	3350	3320
15	Outboard Flywheel Bearing Press.		40.3	41.0	41.3	40.7	40.5	41.1	40.6	41.1
16	Inboard Flywheel Bearing Press.		39.9	40.2	39.5	40.4	39.1	40.0	39.1	39.5
17	Lube Pump Pressure Port Press.		50	50	50	50	50	50	50	50
18	Lube Reservoir Press.		0	0	0	0	0	0	0	0
19	Hyd. Supply Pump Press. at Source		3400	3400	3400	3400	3400	3400	3400	3400
20	Test Motor - Pump Port Pressures (Pumping)	Press.	2900-2700	2890-2650	2860-2760	2940-2890	2900-	2920-2820	2960-2860	2860-2750
21		Return	145-203	150-206	145-200	169-230	164-217	172-226	169-220	163-217
22	Test Motor - Pump Port Pressures (Motoring)	Press.	3300	3305	3280	3350	3320	3320	3340	3317
23		Return	164	167	151	186	181	186	181	181
24	Test Motor - Pump Case Drain Press.		153-208	154-208	153-210	156-210	153-210	156-209	153-210	156-210
25	Vacuum at Flywheel Hub - in. Hg		—	—	—	—	—	—	—	—
26	Vacuum from Hub to Tip (ΔP) in. Hg		—	—	—	—	—	—	—	—
27	Press. at Flywheel Hub - Cm. Hg. Ab.		3.5	3.1	3.5	3.5	3.5	3.5	3.5	3.5
NO.	FLUID FLOW DATA									
28	Hyd. Supply Pump Flow - GPM		18.3	18.3	18.0	19.8	19.6	19.6	18.5	19.2
29	Inlet Flow to Servo - GPM		34.3	34.5	34.5	33.8	35.2	36.2	33.9	30.9
30	Case Drain of Motor - GPM		.3-.55	.35-.55	.3-.55	.3-.55	.33-.55	.35-.55	.3-.55	.3-.55
31	Outb'd. Bearing Lube Flow cc/min		595	595	595	595	595	595	595	595
32	Inb'd. Bearing Lube Flow cc/min		610	610	610	610	610	610	610	610
33	Axial Vibration		14.5	13.9	13.1	12.6	11.3	13.2	12.9	13.5

Table XII (Concluded)

INTERMITTENT DUTY CYCLE SIMULATOR TEST DATA
(HYDRAULIC POWER EXTRACTION)

Data Sheet No. 3

Date			6-20	6-22	6-23	6-23	6-24	6-24		
Accum. Running Time of Test - Hours			92.5	98.3	104.2	110.2	116.2	122.1		
Total Operation Load Cycles			7500	8000	8500	9000	9500	10000		
Vibration Amp. at Flywheel Bearings g's		Inboard	6.4	5.5	5.9	5.7	5.7	5.9		
		Outboard	7.2	7.2	7.2	6.8	7.6	7.2		
Flywheel Speed at Full Motoring - RPM			54500	52500	51600	52400	51600	51150		
Minimum Speed Attain Under Load - RPM			49200	46500	45500	47000	45500	45950		
Average Operational Time Sec./Cycle		Gear Up	5	5	5	5	5	5		
		Gear Down	6	6	6	6	6	6		
Dwell Time - Seconds		Gear Up	13	13	13	13	13	13		
		Gear Down	16-1/2	16-1/2	16-1/2	16-1/2	16-1/2	16-1/2		
Max. Load on Gear - lbs.		Gear Up	5650	5150	5250	5250	5400	5284		
		Gear Down	7620	7030	7270	7240	7400	7270		
NO.	TEMPERATURE DATA - DEG. F									
1	Test Motor - Pump Inlet Temp.		110	110	104	108	105	110		
2	Test Motor - Pump Return Temp.		110	109	106	108	106	110		
3	Test Motor - Pump Case Drain Temp.		146	147	147	141	147	145		
4	Main Hyd. Reservoir Supply Temp.		105	102	102	100	105	105		
5	Gear Box Lube Oil Return Temp.		178	172	168	180	183	163		
6	Gear Box Lube Oil Suction Temp.		74	72	72	70	72	72		
7	Gear Box Scavanger Oil Return Temp.		154	154	156	160	156	158		
8	Gear Box Brg. Adj. Pump Temp.		168	174	173	173	175	170		
9	Gear Box Brg. Adj. Flywheel Temp.		190	192	194	198	194	197		
10	Inboard Flywheel Bearing Temp.		178	178	184	188	181	186		
11	Outboard Flywheel Bearing Temp.		167	176	176	178	170	173		
12	Flywheel Case Housing Temp.		154	150	163	175	152	155		
NO.	SYSTEM PRESSURE DATA - PSIG									
13	Actuator Press - Tension-Gear Up		3250	3270	3265	3210	3240	3240		
14	Actuator Press - Comp.-Gear Down		3360	3365	3360	3320	3270	3250		
15	Outboard Flywheel Bearing Press.		40.9	41.0	41.0	41.5	41.0	41.0		
16	Inboard Flywheel Bearing Press.		39.5	40.0	40.0	40.0	40.0	39.9		
17	Lube Pump Pressure Port Press.		50	50	50	50	49	49		
18	Lube Reservoir Press.		0	0	0	0	0	0		
19	Hyd. Supply Pump Press. at Source		3400	3400	3400	3400	3400	3400		
20	Test Motor - Pump Port Pressures (Pumping)	Press.	2790-2880	2785-2830	2820-2900	2800-2855	2805-2920	2815-2890		
21		Return	163-220	172-226	166-220	169-223	173-224	173-221		
22	Test Motor - Pump Port Pressures (Motoring)	Press.	3320	3315	3320	3330	3310	3325		
23		Return	178	186	183	186	190	185		
24	Test Motor - Pump Case Drain Press.		158-212	156-210	155-210	156-209	157-209	157-210		
25	Vacuum at Flywheel Hub - in. Hg		—	—	—	—	—	—		
26	Vacuum from Hub to Tip (ΔP) in. Hg		—	—	—	—	—	—		
27	Press. at Flywheel Hub - Cm. Hg. Ab.		3.2	3.3	3.2	3.4	3.2	3.2		
NO.	FLUID FLOW DATA									
28	Hyd. Supply Pump Flow - GPM		18.6	14.1	19.4	18.7	19.5	19.6		
29	Inlet Flow to Servo - GPM		31.4	33.4	30.7	30.7	28.0	29.3		
30	Case Drain of Motor - GPM		.3-.55	.3-.55	.3-.55	.3-.55	.3-.55	.3-.55		
31	Outb'd. Bearing Lube Flow cc/min		595	595	595	595	595	595		
32	Inb'd. Bearing Lube Flow cc/min		610	610	610	610	610	610		
33	Axial Vib.		12.9	12.0	14.2	14.2	13.9	13.5		

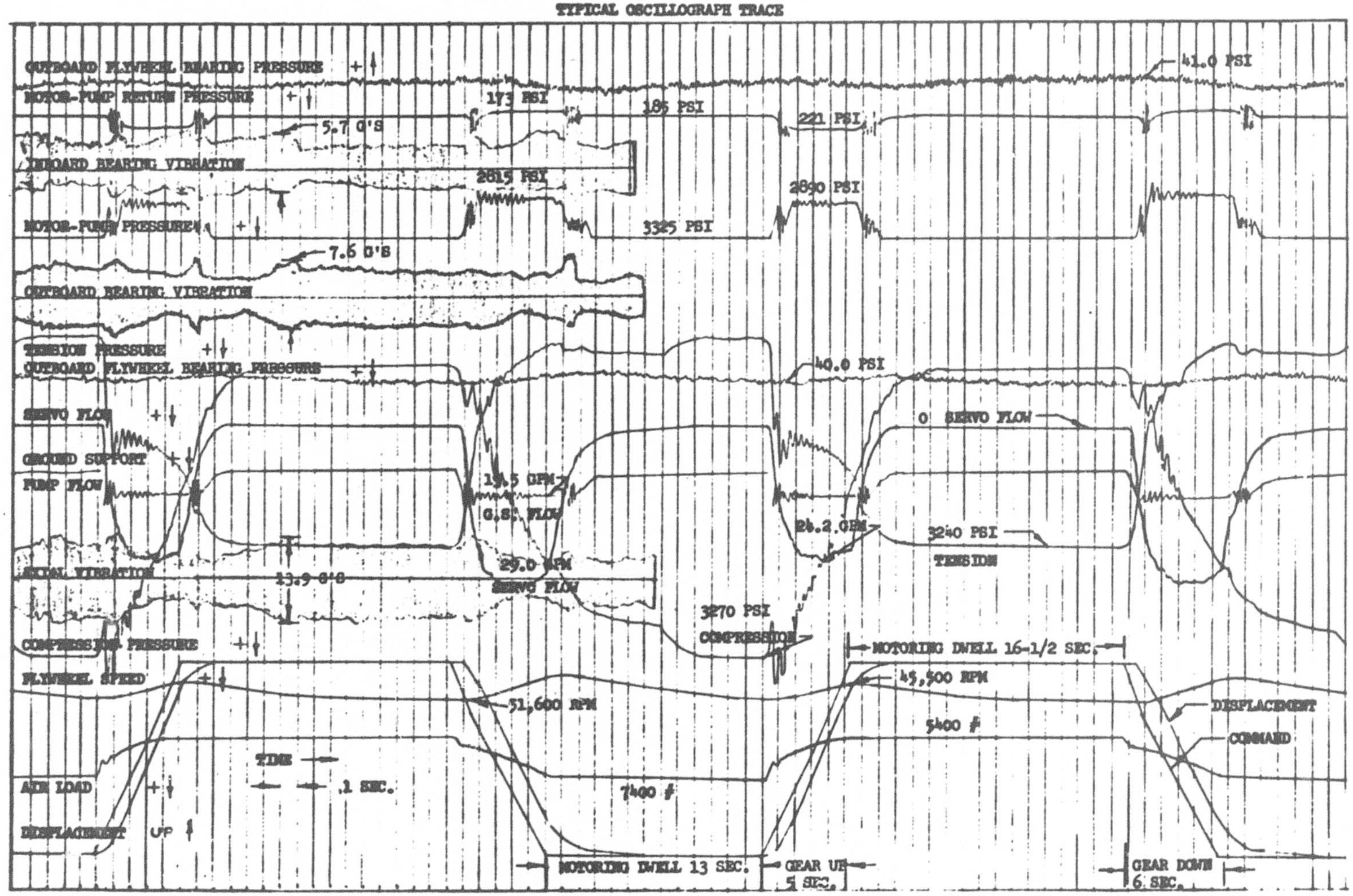


Figure 38. Landing Gear (Hydraulic) Oscillograph Record

Table XIII

INTERMITTENT DUTY CYCLE ENDURANCE TEST RESULTS SUMMARY

a.	Flywheel, gearbox, motor moment of inertia	.268 in.-lb sec^2
b.	Flywheel speed at start of gear down cycle	51,600 rpm*
c.	Flywheel speed at end of gear down cycle	44,600 rpm*
d.	Flywheel speed at start of gear up cycle	49,400 rpm*
e.	Flywheel speed at end of gear up cycle	45,500 rpm*
f.	Flywheel kinetic energy reduction during gear down	990,000 in.-lb
g.	Flywheel kinetic energy reduction during gear up	541,000 in.-lb
h.	Average power supplied by flywheel during gear down	25.0 HP
i.	Average power supplied by flywheel during gear up	16.4 HP
j.	GPU flow during gear down	19.5 gpm
k.	Servo valve flow during gear down	29.0 gpm
l.	Substation flow during gear down	9.5 gpm
m.	Substation pressure differential during gear down	2,642 psi
n.	Substation hydraulic power supplied during gear down	14.65 HP
o.	GPU flow during gear up	19.5 gpm
p.	Servo valve flow during gear up	24.2 gpm
q.	Substation flow during gear up	4.7 gpm
r.	Substation pressure differential during gear up	2,669 psi
s.	Substation hydraulic power supplied during gear up	7.3 HP
t.	Efficiency of power extraction during gear down (n ÷ h) x 100	58.7 percent
u.	Efficiency of power extraction during gear up (s ÷ i) x 100	44.5 percent

* Representative speeds observed during testing

of the motor-pump as a pump, and those of the flywheel and its associated components, i.e., the gearbox, lube pump, and vacuum pump.

It is assumed that the motor-pump when operating at rated pumping conditions (i.e., 3,000 psi, 8,000 rpm and full displacement) is 85 percent efficient. On this basis the fixed losses would compute to be about a 77.5 percent efficiency in the system as it was operated during the pumping phase of the duty cycle, i.e., 9.5 gpm at 2,642 psi and speeds less than 8,000 rpm.

Based on an efficiency of 77.5 percent for the pump, the flywheel losses coupled with the lube pump, vacuum pump, and gearbox would be 6.1 horsepower in the gear-down cycle. A similar computation for the gear-up cycle shows an equivalent 5.1 horsepower loss.

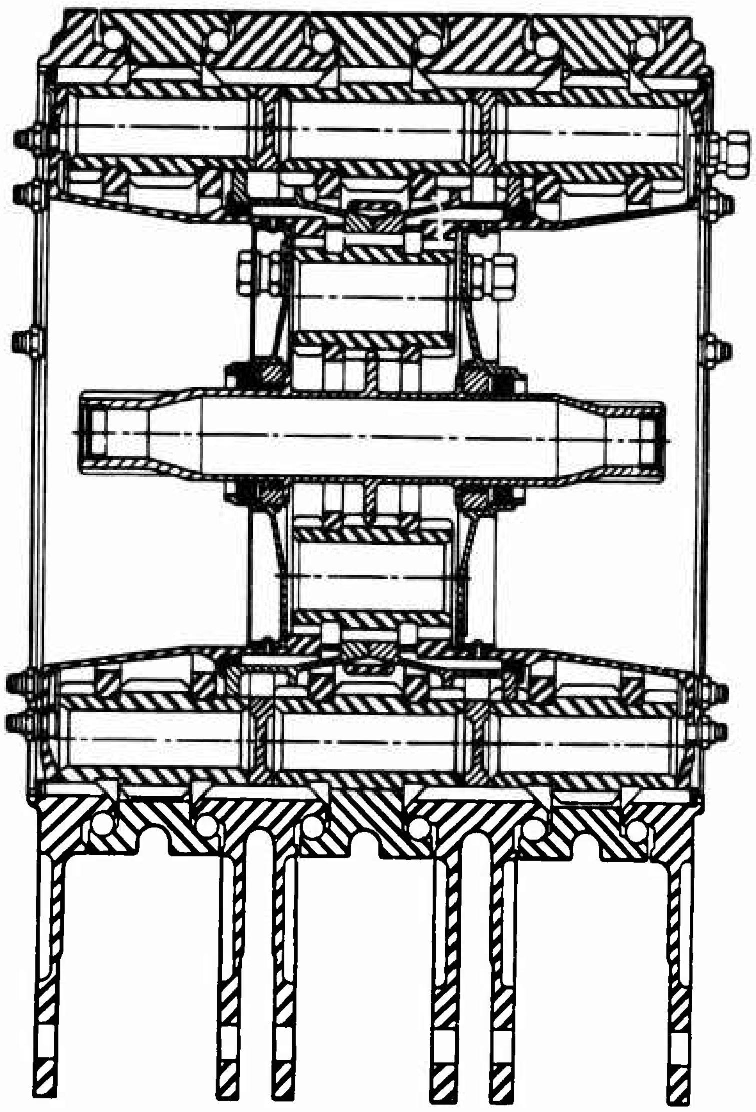
These values of flywheel and associated component losses agree within normal test tolerances with the 5.0 horsepower loss determined from the Tilt Table Endurance Test.

C. MECHANICAL POWER EXTRACTION SYSTEM ENDURANCE TEST

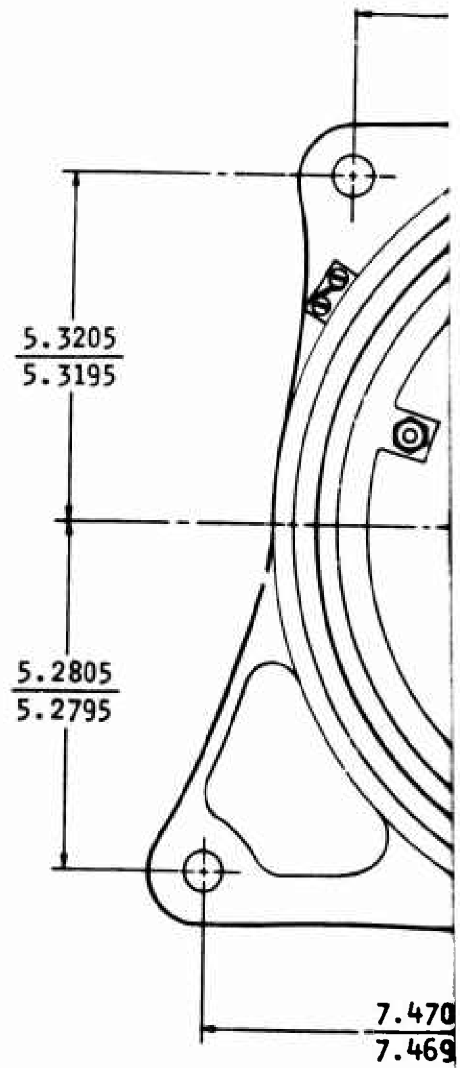
Figures 29, 30, and 31 are applicable to the mechanical power extraction test setup for the Intermittent Duty Cycle Simulator. Additional to this, figures 39 and 40 are drawings of components unique to this arrangement.

The motor-pump used in this test arrangement was the prototype New York Airbrake type described in the section covering the Tilt Table Simulator. It was allowed to operate in the pumping mode only during shutdown of the energy storage substation. All other operation was with full displacement (.4 $in.^3$ rev) as a hydraulic motor using a flow regulator on the hydraulic inlet line to limit speed.

The clutch units, figure 40, are Curtiss-Wright Part Number 164429, Federal Stock Number 1610-786-7817 used in C-133 aircraft propellers. The brake is Curtiss-Wright Part Number 161510, Federal Stock Number 1610-715-0714. The clutch, as supplied, delivered approximately 2,000 inch-pounds lockup torque with a 1,500 inch-pound engagement torque. The clutch and power hinge input shaft speed was set at 3,238 rpm which was nearly six times the speed used in the C-133 propeller application but still tended to slow gear actuation. The present XB-70 Main Landing Gear operating times are 12.5 seconds, extend, 7.5 seconds retract. To meet the 7.5 seconds time with an unmodified hinge would have required that the input gear, which was designed for 400 rpm, must turn at 8,000 rpm. In effect this would have meant that the 52,000 rpm available from the flywheel must be stepped down to 3,238 rpm in the clutches and then up to 8,000 rpm at the hinge input so that the hinge could step it down to 2 rpm. The obvious solution to this problem would have been to revise the power hinge input gearing to eliminate this "step-up". However, to do so would have involved a major revision to the power hinge and a major delay in program completion.



SECTION B-B



A

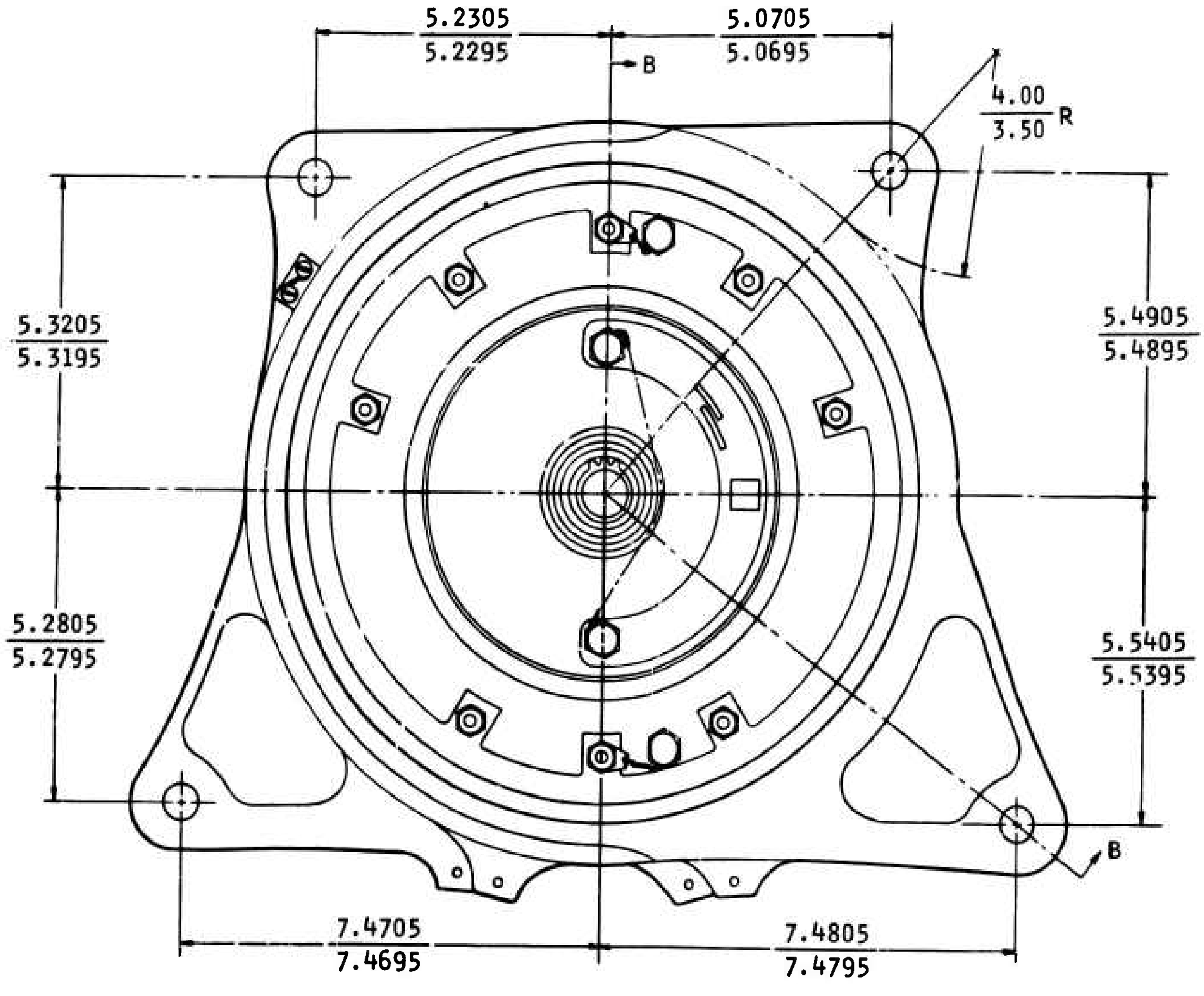
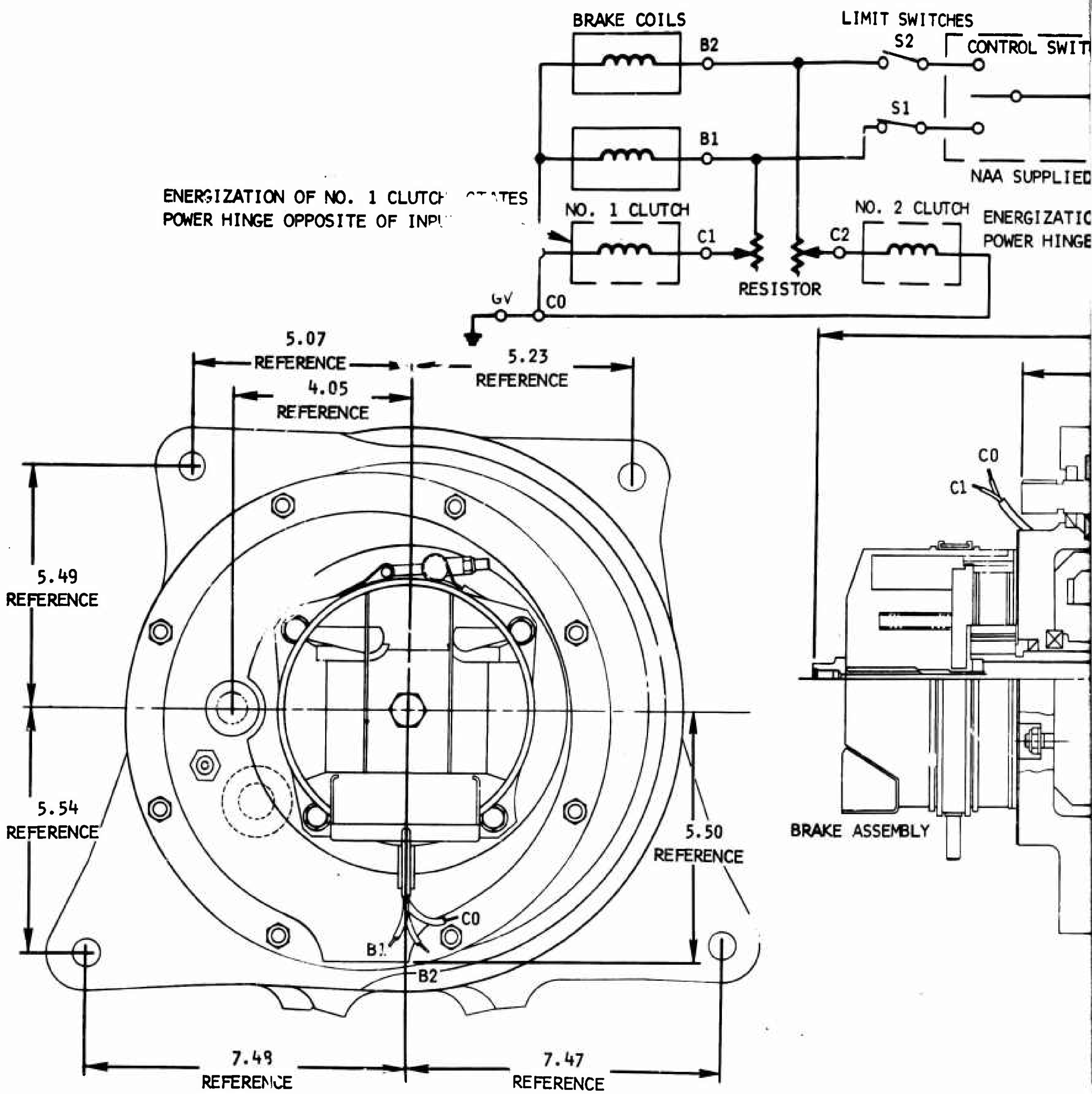


Figure 39. Mechanical Hinge

B


BRAKE COILS
LIMIT SWITCHES
S2
CONTROL SWIT
B2
S1
B1
NAA SUPPLIED
ENERGIZATION OF NO. 1 CLUTCH
POWER HINGE OPPOSITE OF INP
NO. 1 CLUTCH
NO. 2 CLUTCH
ENERGIZATIC
POWER HINGE
C1
C2
RESISTOR
CO
5.07
REFERENCE
5.23
REFERENCE
4.05
REFERENCE
CO
C1
5.49
REFERENCE
5.54
REFERENCE
5.50
REFERENCE
BRAKE ASSEMBLY
CO
B2
7.47
REFERENCE
REFERENCE

A

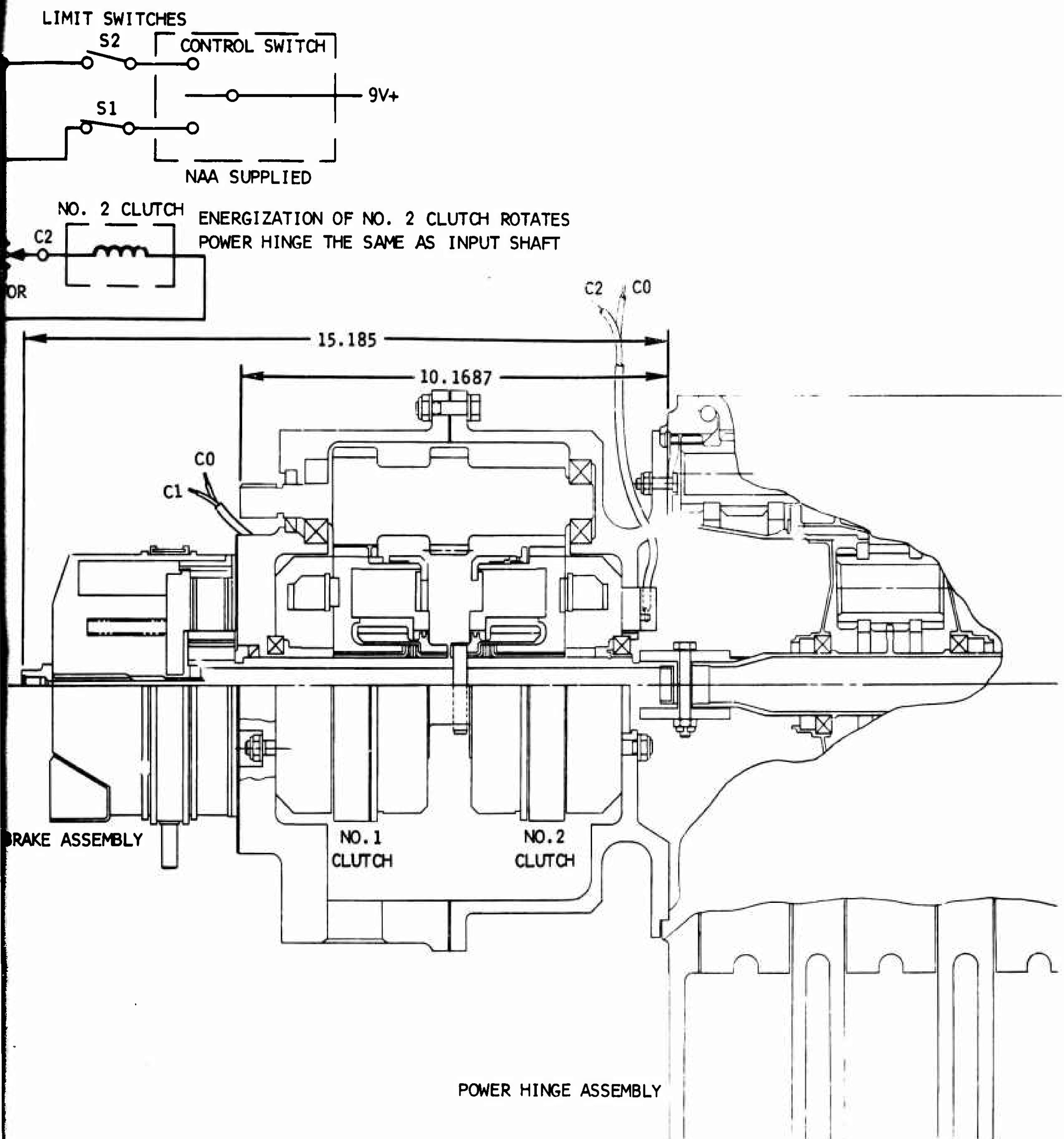


Figure 40. Clutch and Brake Assembly

B

A uniform gear extend and retract time of 15 seconds was selected. This represented a deviation from actual air vehicle duty cycles and actual air vehicle gear operation. On closer examination, however, these operating times were considered satisfactory for the following reasons:

1. Irrespective of operating time the output hinge moments remained unchanged.

2. The energy extracted from the flywheel per cycle remained essentially unchanged as a result of changed cycle time.

3. The peak torque experienced by the flywheel was a function of clutch slip-torque characteristics and was independent of operating time.

4. The peak power extraction was reduced by a factor of approximately two. However, since the fatigue stresses in the flywheel, clutch, and gearbox were primarily a function of torque and not of power, this factor did not have a significant effect on the validity of the tests.

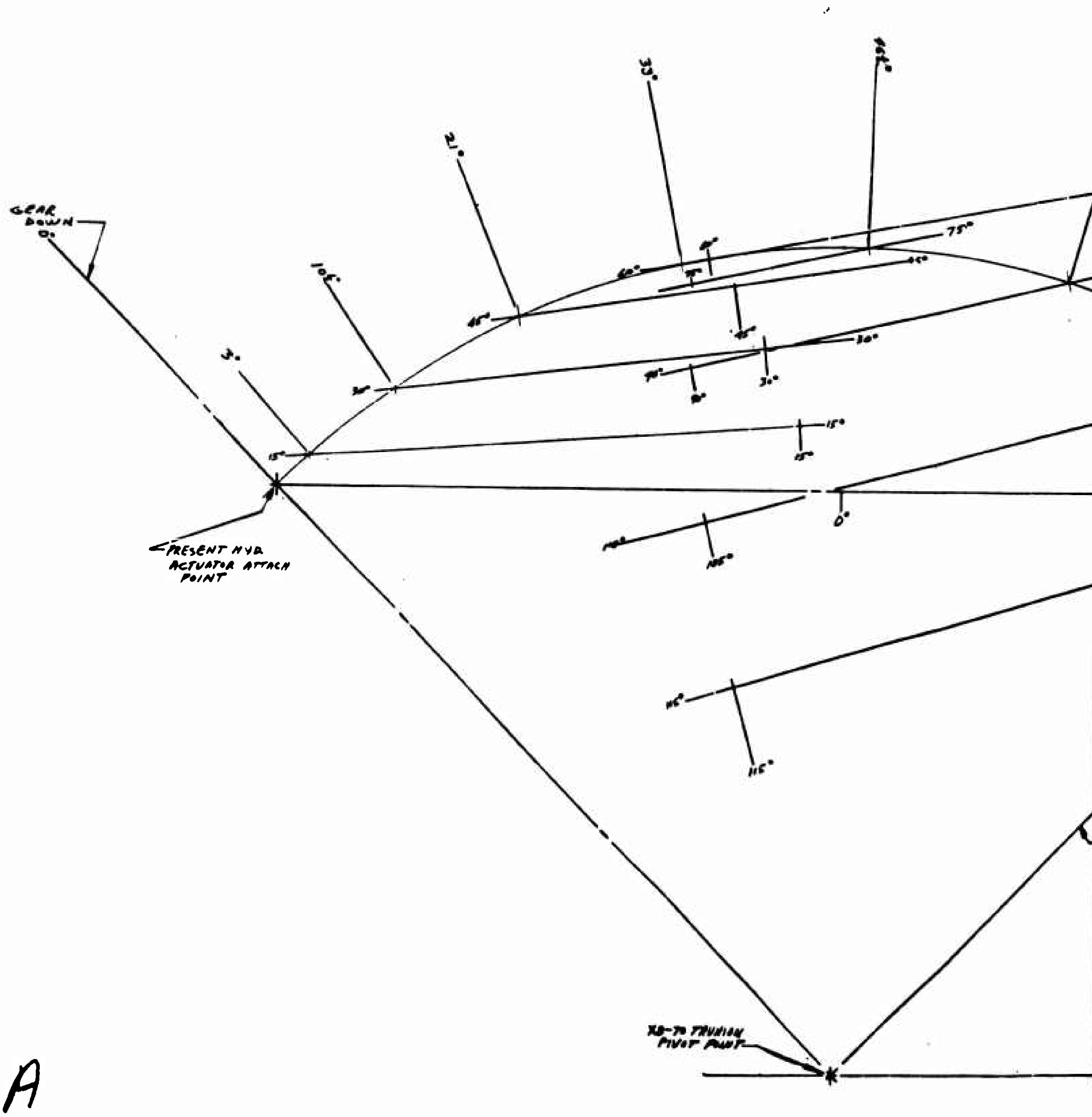
A kinematic arrangement was chosen to connect the landing gear to the power hinge wherein the gear retracted 90 degrees while the hinge rotated 115 degrees. This is shown as figure 41. The hinge was overcenter when the landing gear was extended.

This arrangement permited the clutches to engage with aiding loads (extend) or no-load (except inertia of the power hinge input shaft) (retract). It permitted the brake to engage with opposing load on the system (upstop) or with no-load originating from the landing gear (down-stop over-center). It was believed that this approach was very close to optimum for this landing gear configuration.

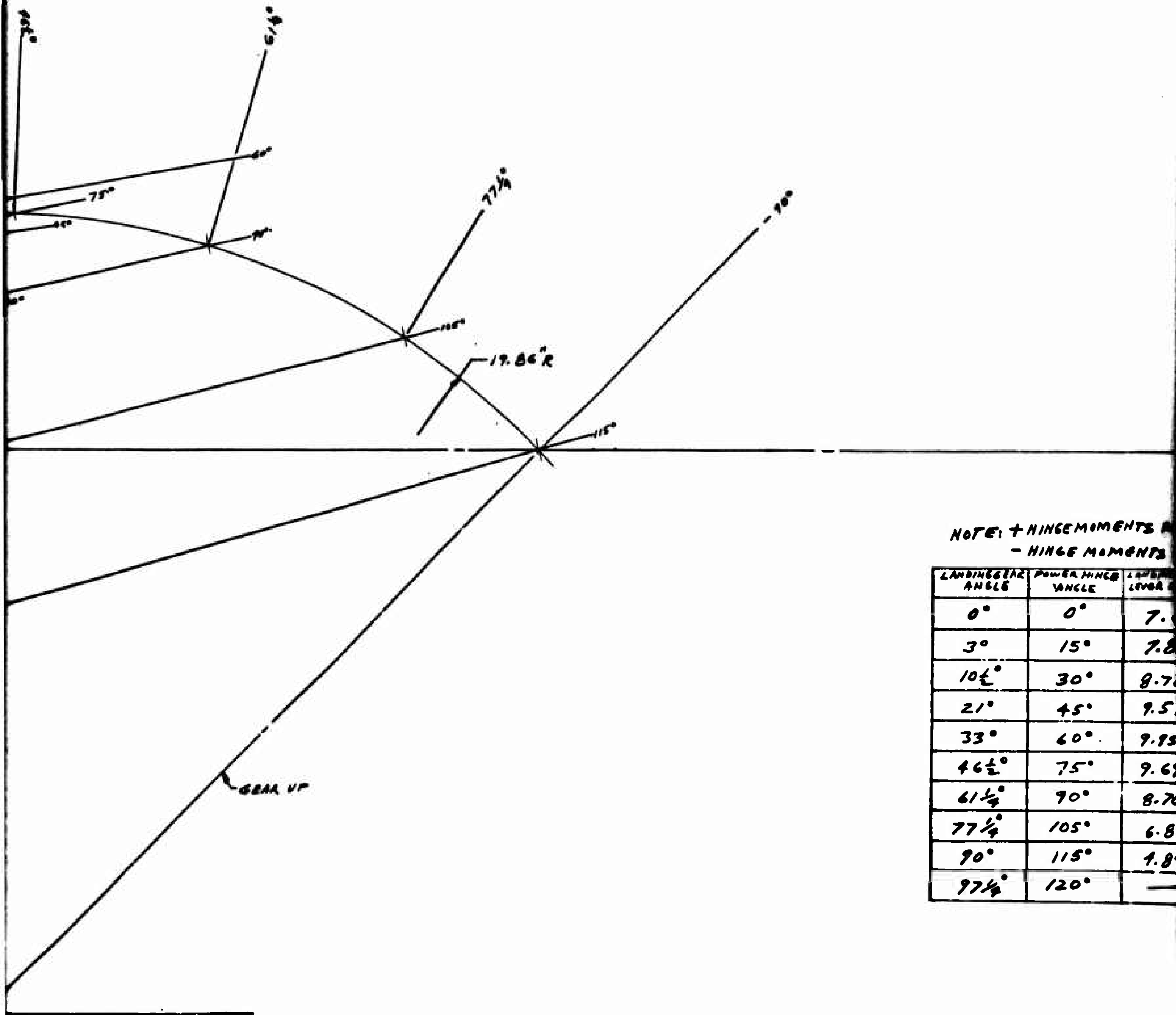
The control for this system was connected to an automatic timer wherein a switch was closed to one position and the brake and one of the two clutches was energized. This released the brake and engaged the energized clutch. At the end of the actuation stroke a limit switch would break the electric circuit, engaging the brake and disengaging the clutch. A similar sequence occurred in the reverse direction for the opposite direction of landing gear actuation.

When the endurance test was started, a total of 382 cycles were completed with no noticeable irregularities. The clutch for engaging gear-down then began to slip at the maximum load portion of the cycle and overheated to the extent that the clutch plates became warped and could not be separated from each other in the space that normally remains for the de-energized clutch plate stack. The exact cause for this failure was not immediately established, even though an extensive effort was made to analyze the details of operation associated with the test rig and test specimen.

It was noted during refurbishment that the gear-up clutch plates showed signs of overtemperature, even though this assembly was functioning properly.


GEAR DOWN 0°
PRESENT HYD ACTUATOR ATTACH POINT
PIVOT POINT
3°
10½°
21°
33°
75°
30°
15°
0°

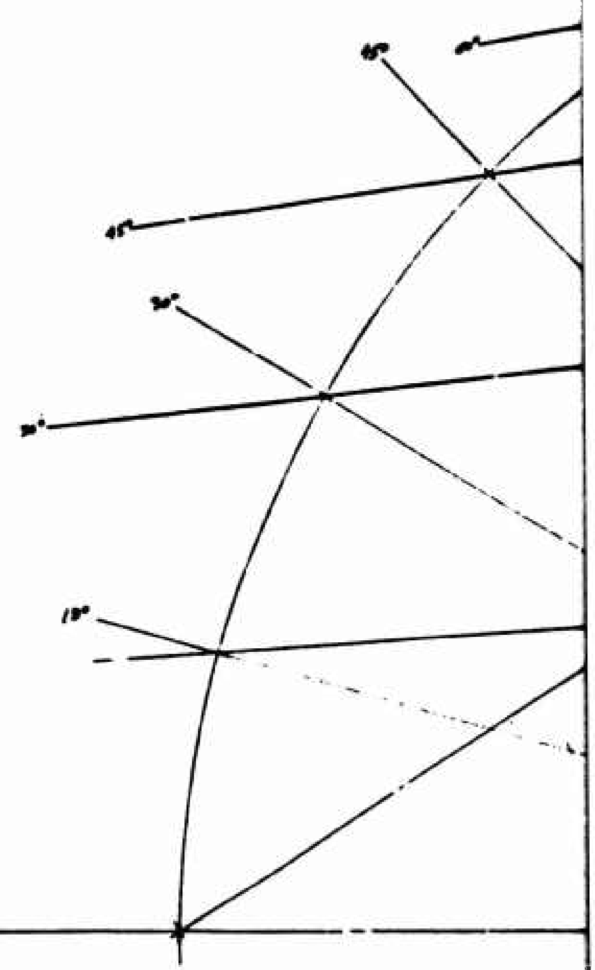
A



NOTE: + HINGEMOMENTS
- HINGE MOMENTS

LANDINGGEAR ANGLE	POWER HINGE ANGLE	LEVER
0°	0°	7.
3°	15°	7.[illegible]
10½°	30°	8.7
21°	45°	9.5
33°	60°	9.95
46½°	75°	9.6
61¼°	90°	8.7
77¼°	105°	6.8
90°	115°	1.8
97¼°	120°	—

B



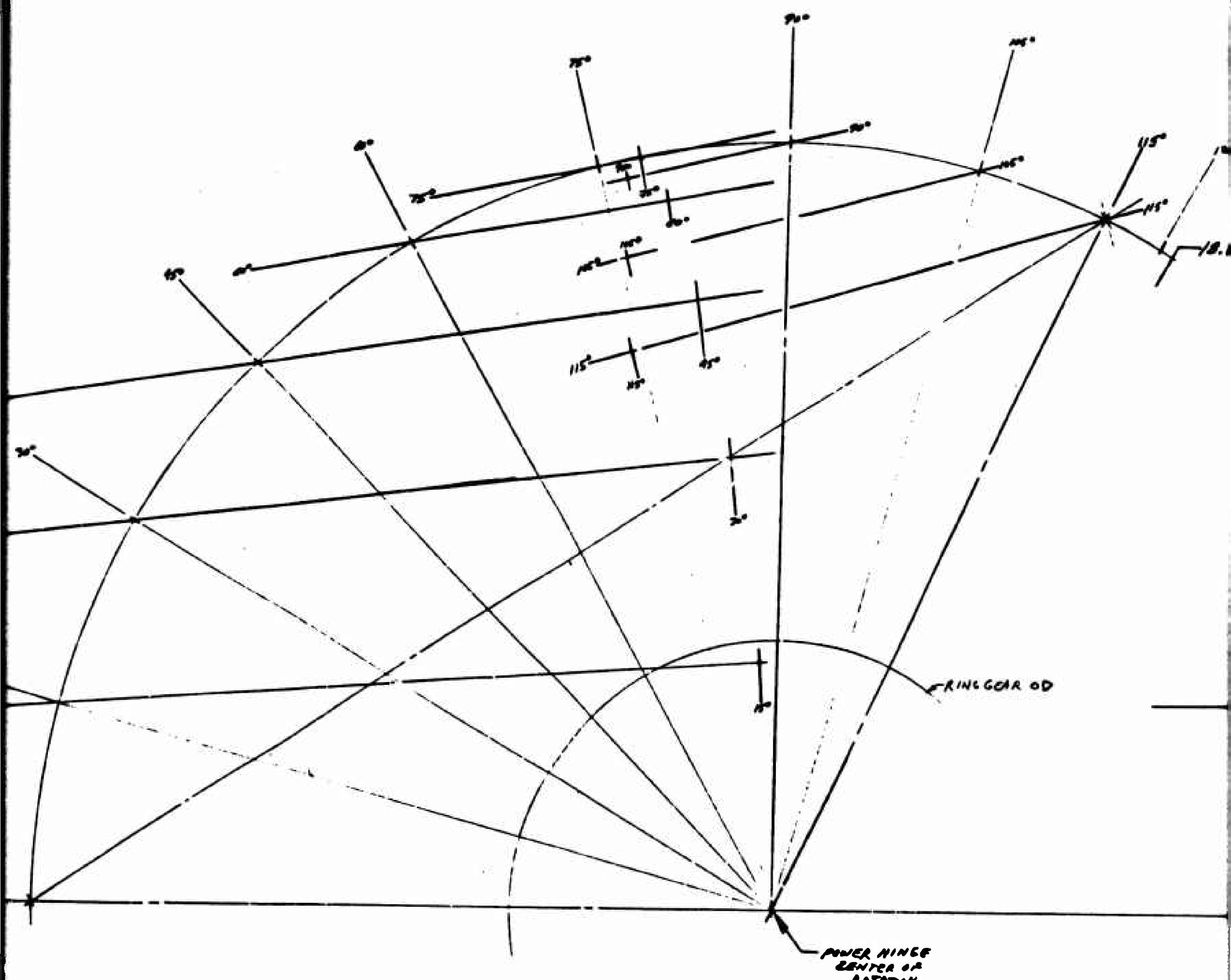
NOTE: + HINGE MOMENTS RESIST MOTION
— HINGE MOMENTS AID MOTION

MOTION

LANDING GEAR ANGLE	POWER HINGE ANGLE	LANDING GEAR LEVER ARM	POWER HINGE LEVER ARM	LANDING GEAR HINGE MOMENT RETRACTING	LANDING GEAR HINGE MOMENT EXTENDING	LEVER ARM RATIO ($\frac{\text{POWER HINGE}}{\text{LANDING GEAR}}$)	POWER HINGE HINGE MOMENT RETRACTING	POWER HINGE HINGE MOMENT EXTENDING
0°	0°	7.00	0.0	−1,083,259	+930,000	$\frac{0.0}{7.0} = 0$	0	0
3°	15°	7.82	2.93	−965,000	+920,000	$\frac{2.93}{7.82} = .375$	−362,000 IN-LB	+345,000 IN-LB
$10\frac{1}{2}$°	30°	8.78	5.37	−900,000	+875,000	$\frac{5.37}{8.78} = .611$	−550,000 IN-LB	+535,000 IN-LB
21°	45°	9.57	7.27	−805,000	+820,000	$\frac{7.27}{9.57} = .760$	−611,000 IN-LB	+623,000 IN-LB
33°	60°	9.95	8.48	−700,000	+725,000	$\frac{8.48}{9.95} = .853$	−597,000 IN-LB	+618,000 IN-LB
$46\frac{1}{2}$°	75°	9.69	9.05	−480,000	+500,000	$\frac{9.05}{9.69} = .935$	−449,000 IN-LB	+467,000 IN-LB
$61\frac{1}{4}$°	90°	8.70	8.85	−40,000	+75,000	$\frac{8.85}{8.70} = 1.02$	−40,800 IN-LB	+76,500 IN-LB
$77\frac{1}{4}$°	105°	6.85	7.93	+375,000	−340,000	$\frac{7.93}{6.85} = 1.16$	+435,000 IN-LB	−394,000 IN-LB
90°	115°	4.89	6.83	+513,000	−430,000	$\frac{6.83}{4.89} = 1.41$	+723,000 IN-LB	−606,000 IN-LB
$97\frac{1}{4}$°	120°	—	—	—	—	—		

MOTION

C



MOTION

HINGE RACTING	POWER HINGE HINGE MOMENT EXTENDING
	0
W-LB	+345,000 IN-LB
IN-LB	+535,000 IN-LB
N-LB	+623,000 IN-LB
-LB	+618,000 IN-LB
IN-LB	+467,000 IN-LB
IN-LB	+76,500 IN-LB
IN-LB	-379,000 IN-LB
N-LB	-606,000 IN-LB

MOTION

D

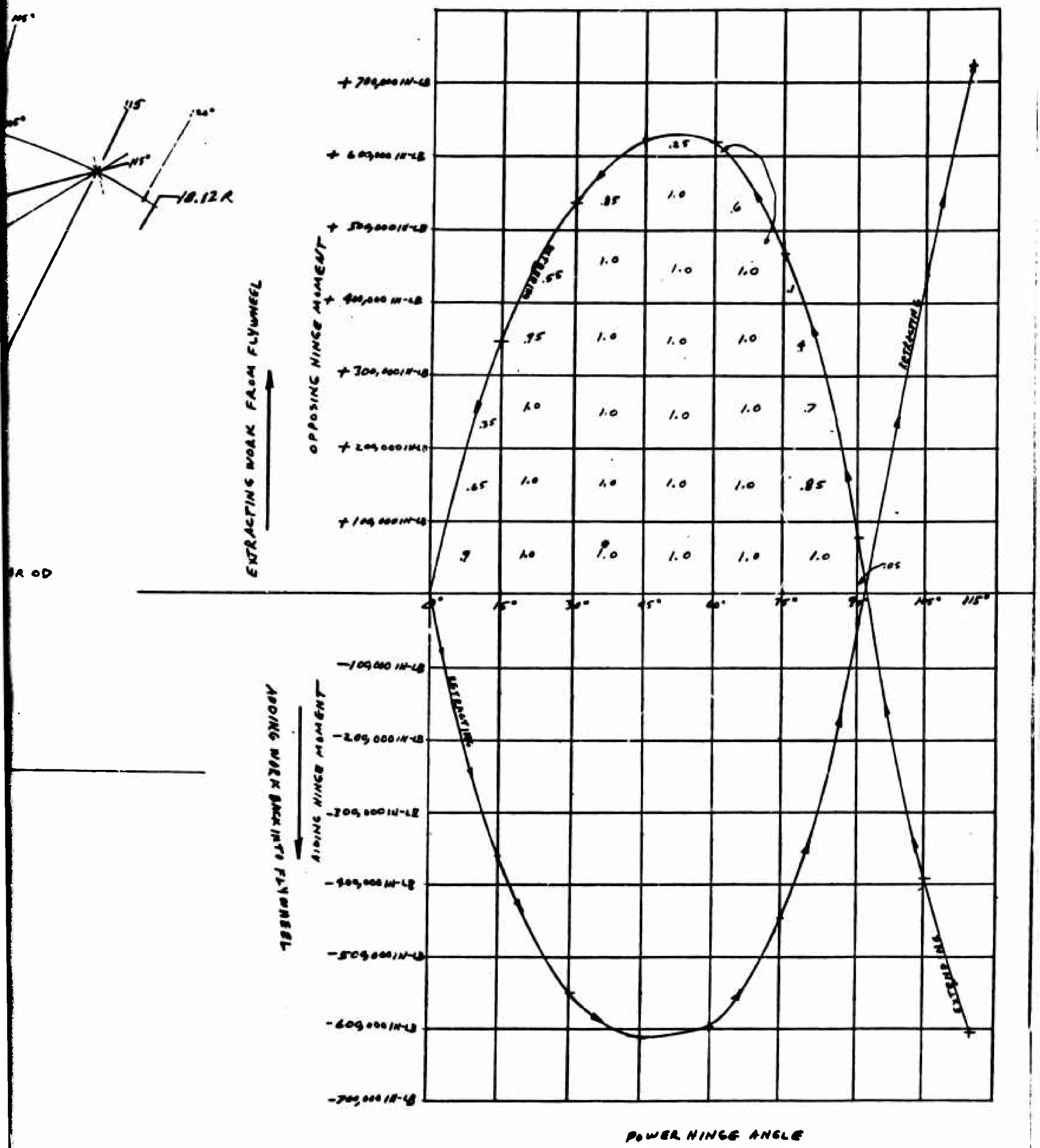


Figure 41. Landing Gear Kinematics

E

A number of potential causes, which were considered, were as follows:

1. Interruption of the clutch lubrication and cooling oil supply

2. Inadvertent energization of the gear-up clutch during the gear-down cycle

3. Inadvertent de-energization of the brake during the gear-down cycle

4. Overload of the clutch by improper operation of the simulated airload system, or by binding in the bearings which attach the linkages to the strut

5. Inadvertent partial de-energization of the gear-down clutch during the gear-down cycle

Following completion of the refurbishment of the clutches, the unit was reinstalled on the simulator. All systems and equipment were checked for proper operation, and endurance testing was resumed.

On the maximum load portion of the second gear-down cycle a repeat of the first failure occurred. In this case the clutch bindup was sufficient to bring the flywheel to zero rpm in about 5 seconds.

The unit was removed and was again returned to the supplier. Disassembly and inspection revealed that the gear-down clutch had done some slipping and showed signs of overtemperature. However, the location of the principal drag loss, where maximum heating occurred and where the flywheel energy must have been consumed with such rapidity, was in the gear-up clutch plates. With this in mind, and remembering that the relative slipping speed of the de-energized clutch doubles during a gear-up or gear-down operation, two additional causes for failure were advanced as possibilities.

1. Improper proportioning, distribution or inadequate quantity of cooling oil supply to the clutches

2. Improper configuration of the clutch plate stack with respect to diametral size, rotational speed, quantity of plates, and space between plates while operating in a de-energized condition

The assembly was completely refurbished and modifications made to increase the space between plates as much as possible within the confines of the existing envelope as well as modifications to direct and more evenly distribute a larger amount of the cooling oil to the plates. These changes were deemed prudent since as has already been pointed out, this application of the clutch calls for speeds several times that for which it was originally designed. Any additional changes of still larger scope which might have been necessary to correct the problem, such as a reduction of the clutch plate diameter, or a change in the clutch speed, through gearing revisions, were not considered possible within the effort and schedule limits of the program.

The assembly was then reinstalled on the simulator and the test again resumed. Performance was satisfactory; however, as the test progressed it was noted that the dwell time between cycles needed to be extended to permit the flywheel to regain its upper operating speed. Additionally the hydraulic supply pressure to the motor had to be progressively increased. Degradation was also noted with respect to an increase in the speed reduction experienced during the gear-down cycle.

After reaching a point somewhere in excess of 1,650 cycles a slipping of the gear-down clutch began to occur intermittently during the heaviest load portion of the stroke. When this occurred, it became standard procedure for the operator to stop the actuation and reverse the stroke. The opposing airloads then became aiding airloads and the gear could be retracted. After this reversal, normal cycles could be operated. However, as would be expected, the frequency of occurrence of this slipping increased, and eventually required shutdown of the simulator at 1,745 cycles.

After a review of the problems, the corrective action previously taken, and test data accumulated to date, it was apparent that a major redesign would be necessary to accomplish the original number of test cycles without an extensive number of refurbishments; therefore, testing was terminated.

The test did, however, accomplish the basic objective of demonstrating the mechanical power extraction approach for energy storage substations, even though the components were misconfigured for this application. Additional testing would have served no useful purpose other than to show 10,000 cycles could be attained after repeated refurbishment of this hardware, where in reality a proper design would require components especially constructed for this application.

Data from this test are shown in table XIV. Figure 42 is a data record taken near the end of the test.

From the data, table XV was created to summarize the performance. The efficiency of 18.7 percent for the mechanical system during gear-down would, on the surface, seem unattractively low; however, this number represents the inefficiency of all of the elements in series, from the flywheel to the airload cable in one path, and from the hydraulic motor to the airload cable in another path. A breakdown of the various elements in these paths is not possible because there were no dynamic torque measurements recorded.

A similar figure in the hydraulic power extraction system would use 25 horsepower as flywheel input, 30 horsepower as GPU input resulting in a total of 2,180,000 inch-pounds over a 6-second time span. This would represent 32.7 percent efficiency including all of the elements in the load path to the airload cable.

Before the test, it was believed that the mechanical system would show a much higher efficiency than the hydraulic power extraction system. The reason that it did not is principally because of the no-load losses that

Table XIV

INTERMITTENT DUTY CYCLE SIMULATOR TEST DATA (MECHANICAL POWER EXTRACTION)

Summary Data Sheet No.

No.	Item								
	DATE		7-31	8-1	8-10	9-4	9-6	9-7	9-7
	Accum. Running Time of Test - Hours		127.3	131.1	136.8	144.4	150.6	156.4	162.3
	Total Operation Load Cycles		50	200	500	750	1000	1250	1500
	Vibration Amp. at Flywheel Bearings g's	Inboard	-	9.8	8.1	8.1			
		Outboard	9.1	11.7	7.9	13.6	13.6	12.0	8.4
		Axial	15.8	16.7	14.2	16.8	15.5	14.5	15.5
	Flywheel Speed at Full Motoring - RPM		49,600	49,800	51,000	49,000	52,400	49,500	49,400
	Minimum Speed Attain Under Load - RPM		39,200	38,800	45,000	37,200	35,700	38,700	37,000
	Average Operational Time Sec./Cycle	Gear Up	20	18	15	15	16	15	15
		Gear Down	20	16	17	15	18	18	18
	Dwell Time - Seconds	Gear Up	25	32	17	17	15	15	16
		Gear Down	25	40	30	33	32	30	31
	Air Load on Gear - lbs.	Gear Up	5280	5560	5740	5170	5500	5660	5340
		Gear Down	6795	6780	7490	6840	7240	7510	7000
	Push-Pull Arm Load - Total	Gear Up	+56,600	56,800	60,000	61,300	63,200	62,600	63,000
	Push-Pull Arm Load - Total	Gear Down	-65,900	-60,300	-70,600	-66,400	-67,200	-71,000	-64,600
NO.	TEMPERATURE DATA - DEG. F								
1	Test Motor - Pump Inlet Temp.		99	105	110	95	99	100	100
2	Test Motor - Pump Return Temp.		102	107	107	93	95	97	96
3	Test Motor - Pump Case Drain Temp.		120	137	160	146	150	154	153
4	Main Hyd. Reservoir Supply Temp.		100	108	105	98	100	101	100
5	Gear Box Lube Oil Return Temp.		103	108	210	173	172	183	173
6	Gear Box Lube Oil Suction Temp.		95	76	83	65	66	70	63
7	Gear Box Scavenger Oil Return Temp.		122	160	168	142	149	163	156
8	Gear Box Brg. Adj. Pump Temp.		137	165	172	162	170	172	173
9	Gear Box Brg. Adj. Flywheel Temp.		142	195	186	158	168	179	174
10	Inboard Flywheel Bearing Temp.		152	185	170	136	145	156	150
11	Outboard Flywheel Bearing Temp.		153	182	190	155	155	162	154
12	Flywheel Case Housing Temp.		153	173	158	90	90	114	95
13	Lube Oil to Clutch - Brake Temp.		-	90	110	108	105	110	108
14	Lube Oil from Clutch - Brake Temp.		-	105	120	115	110	118	110
15	L.H. Side Power Hinge Temp.		250	175	-	-	-	-	-
16	R.H. Side Power Hinge Temp.		230	165	170	160	180	128	200
16a	Lube Oil from Brake Temp.		-	105	90	85	90	125	122-159
NO.	SYSTEM PRESSURE DATA - PSIG								
17	Outboard Flywheel Bearing Press.		39.0	43.4	40.4	28.2	27.4	26.7	25.9
18	Inboard Flywheel Bearing Press.		39.0	43.1	40.5	26.3	26.7	27.7	26.3
19	Lube Pump Pressure Port Press.		50	50	50	48	48	46	48
20	Lube Reservoir Press.		0	0	0	0	0	0	0
21	Hyd. Supply Pump Press. at Source		2960	3100	3300	3335	3400	3420	3440
22	Test Motor - Pump Port Pressure (Motoring)	Press.	2850	2975	3250	3230	3350	3350	3350
23		Return	168.0	168.0	164.5	164.5	164.5	170	170
24	Test Motor - Pump Case Drain Press.		166	168	168	167	167	163-168	163-166
25	Press. at Flywheel Tip - Cm Hg Ab		3.0	3.0	2.5	1.5	1.9	2.2	2.4
26	Lube Press to Clutch		26	25	-	34.0	32.0	32.5	32.0
NO.	FLUID FLOW DATA								
27	Hyd. Supply Pump Flow - GPM		16	16.1	16	16	16	16	17
28	Lube Flow - GPM		-	3.5	4.5	5.0	5.1	5.1	5.0
29	Case Drain of Motor - GPM		-	.33	.45-.52	.45-.52	.50	.47	.37-.48
30	Outb'd. Bearing Lube - Flow cc/min.		545	580	555	685	670	655	640
31	Inb'd. Bearing Lube - Flow cc/min.		560	590	570	705	715	730	705
NO.	POWER SUPPLY								
32	Clutch Power Supply - Voltage - Down				12.5		13.0V	13.0V	12.9V
33	Clutch Power Supply - Voltage - Up				12.0		12.5V	12.5V	12.4V
34	Clutch Power Supply - Amp - Down				8.5		8.9A	8.8A	8.8A
35	Clutch Power Supply - Amp - Up				10.5		11.0A	11.0A	10.5A

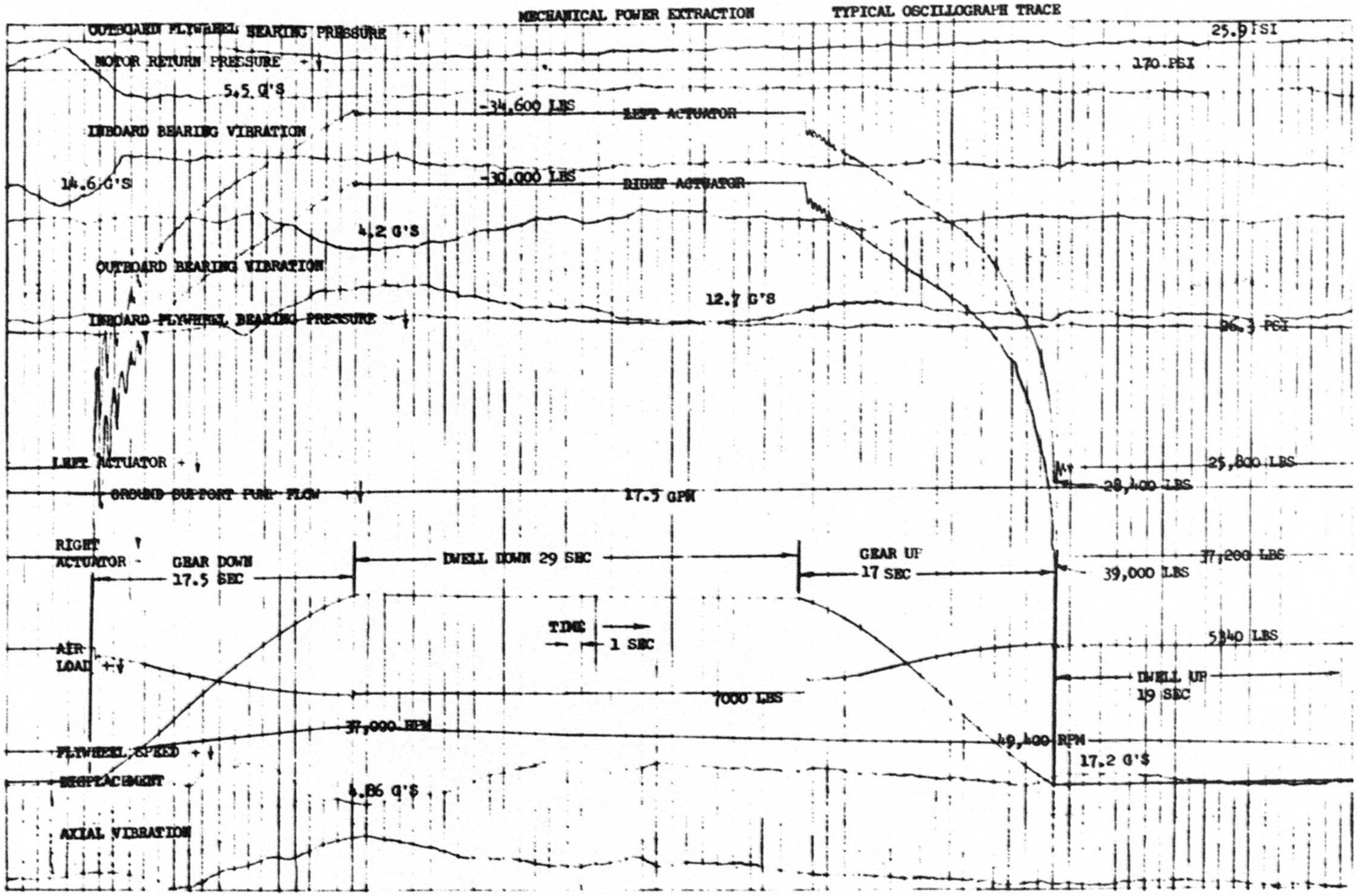


Figure 42. Landing Gear (Mechanical) Oscillograph Record

Table XV

MECHANICAL POWER EXTRACTION ENDURANCE TEST RESULTS SUMMARY

a.	Flywheel, etc, moment of inertia	.268 in.-lb sec^2
*b.	Flywheel speed at start of gear-down cycle	49,600 rpm
*c.	Flywheel speed at end of gear-down cycle	39,200 rpm
d.	Flywheel kinetic energy reduction during gear-down	1,365,000 in.-lb
e.	Average power supplied by flywheel during gear-down	11.7 HP
f.	Hydraulic motor pressure differential	2,680 psi
g.	Hydraulic motor flow	16.0 gpm
h.	Kinetic energy supplied by hydraulic motor during gear-down (85 percent eff)	1,815,000 in.-lb
i.	Average power supplied by hydraulic motor during gear-down	21.2 HP
j.	Total kinetic energy supplied to mechanical system during gear-down	3,180,000 in.-lb
k.	Total average power supplied to mechanical system during gear-down	32.9
l.	Theoretical mechanical energy required by airload and weight	712,042 in.-lb
m.	Theoretical average mechanical power required by airload and weight	5.4 HP
n.	Efficiency of mechanical system during gear-down	22.4 percent

* NOTE The net flywheel speed change during gear-up was frequently equal to zero resulting in no change to the flywheel kinetic energy.

existed in the deenergized clutch. This first became apparent when substation operational checkout required an unusually long time to bring the flywheel up to speed. As noted in the endurance test description, these losses were the result of improper clutch configuration for this application, and eventually lead to early termination of the test.

The flywheel both speeded up and slowed down during the gear-up portion of the cycle; however, in general the net change in speed was for all practical purposes equal to zero. Considering the inefficiencies just cited in the mechanical system, this characteristic of the simulator would indicate that a large amount of the aiding load power was being constructively used to offset losses. It is conceivable that in a properly configured system, the aiding loads of a duty cycle could be used to help restore flywheel speed. This consideration is not applicable in the hydraulic power extraction system.

D. INTERMITTENT DUTY CYCLE SIMULATOR TEST CONCLUSIONS

The tests conducted on the Intermittent Duty Cycle Simulator indicated that mechanical power extraction employing disc clutches and mechanical hinges is very feasible. However, the tests also indicated that, wherever a high inertia power source (such as the flywheel) is connected to, and operates, a high inertia load (such as the landing gear) by means of a clutch, the starting (acceleration) and stopping (deceleration) of the load should be accomplished through overcenter linkage or its equivalent.

As has been previously indicated it was also concluded from these tests that the selection of the clutch size and operating speed is a major factor in the effectiveness of this approach. For this particular application it was concluded that the clutch brake unit should have operated at approximately one-half the speed necessitated by this application (i.e., 1,600 rpm versus this application's 3,238 rpm) and the mechanical hinge's gear reduction ratio should have been reduced to one-third of its present value (i.e., 830:1 versus this application's 2,500:1).

The incorporation of all of these changes (i.e., overcenter linkage, reduced clutch speed and reduced hinge ratio) should lead to numerous improvements in the system as follows:

1. Parasite losses should reduce by 50 percent. This would mean that item k in table XV would become 19.25[33.3 HP - (33.3 - 6.2).50 = 19.25], thus improving the efficiency of item n to 32 percent. It should be noted that this efficiency is nearly a standoff with the equivalent hydraulic system efficiency (32.7 percent) mentioned earlier. However, since the hydraulic system's efficiency is essentially the same for gear-up and gear-down while the mechanical system can take advantage of aiding loads and does not extract energy from the flywheel during gear-up, it may be safely concluded that the mechanical system's cycle efficiency in terms of losses is approximately twice as good as that of the hydraulic system.

2. Heat rejection, which was a problem throughout the mechanical demonstration, would be reduced by 50 percent and would cease to be a significant consideration.

3. Gear cycle time would be comparable to that exhibited by the hydraulic type (i.e., less than 12.5 seconds).

4. Mechanical power extraction component weight would be reduced and would total to about 250 pounds. The weight of the system as tested was 299 pounds and consisted of the following component weights:

Clutch brake unit		102.5 pounds
Mechanical hinge		199.5 pounds
PTO shaft		2.0 pounds
	Total	299.0 pounds

With overcenter linkage the size of the brake could be greatly reduced. This plus refinements in the clutches and gearing should make possible a 33-pound weight reduction in the clutch brake unit. The reduction in mechanical hinge gearing ratio would make possible the revision of the inner stage gearing which in turn would reduce the hinge weight by about 16 pounds.

The tabulated weight data for the system would then become:

Clutch brake unit		69.5 pounds
Mechanical hinge		178.5 pounds
PTO shaft		2.0 pounds
	Total	250.0 pounds

It was concluded when comparing mechanical versus hydraulic power extraction that each had certain basic characteristics which, under a given set of circumstances, might favor its use. The hydraulic system has high efficiency under high load but low efficiency under low load, and no load losses are about as large as the useful work accomplished during peak load. As previously stated, the hydraulic system cannot take advantage of aiding loads. In the presence of aiding loads the flywheel still loses speed. On the other hand, the mechanical systems can store energy in the presence of aiding loads but suffers relatively high energy extraction rates from the flywheel during opposing load portions of the cycle. From this it was concluded that, where load conditions were the primary factor considered, hydraulic systems would tend to be favored in the presence of sustained high opposing loads whereas mechanical systems would be preferred in the presence of moderate average loads and/or reversing loads.

It was also concluded that the mechanical system is inherently more flexible in application. By definition the means for energy input and extraction in the hydraulic system is limited to hydraulic devices (i.e., pumps, motors, and/or pump-motors). Conversely the mechanical system, although limited to mechanical devices for energy extraction, is adaptable to a variety of input devices. These would include hydraulic motors, electric motors, shafting, and gas turbines.

Section V

CONTINUOUS DUTY CYCLE SIMULATOR

A. FLIGHT CONTROL APPLICATION

The Continuous Duty Cycle Simulator was a test arrangement of a simulated flight control system that demonstrated an application of a flywheel energy storage substation. Flight control functions are typified by an almost continuous demand for power at low level (usually below 10 percent of maximum rated power) interspersed with short term power demands approaching 100 percent rated power. The substation in this application was designed to supply a third of the power requirement when the 100 percent rated demand was called for by the duty cycle. On an actual airplane, this would allow the normal engine-driven secondary power system to be designed to slightly larger than continuous duty cycle requirement. Specifically, this simulator was designed to demonstrate the ability of an energy storage substation to supply power to a load bank which simulated the duty cycle and power requirement equivalent to a single system operation of three panels of the XB-70 elevon system.

B. SIMULATOR

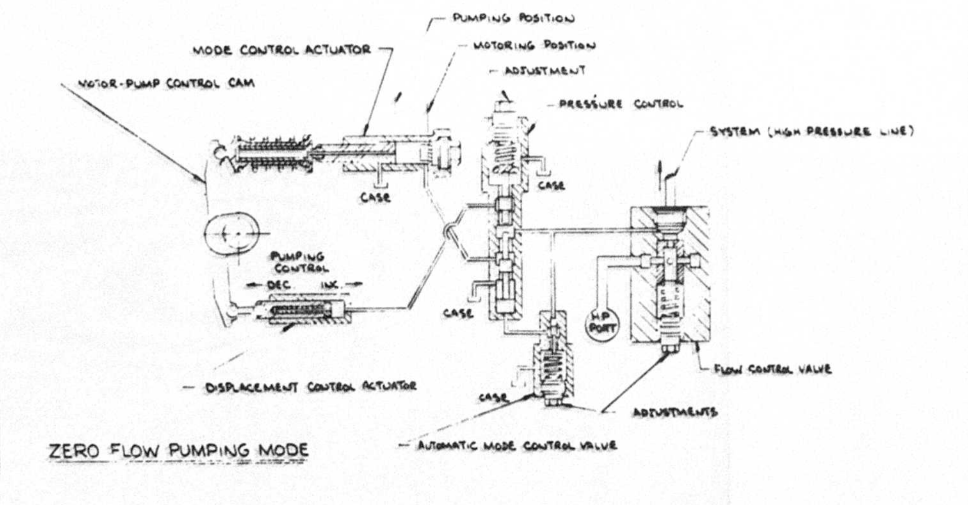
As shown in figure 43, the simulator consisted of a laboratory hydraulic power supply package, a motor-pump hydraulic unit, a gearbox for exchanging energy between the motor-pump and flywheel, a flywheel together with its housing, and a load bank package. The XB-70 elevon panel actuator package was simulated by a linear actuator with equal extending and retracting piston areas which converted flow into linear motion. The actuator operated a linear potentiometer which supplied information for the data record.

The energy storage substation for the simulator used the automatic New York Airbrake type motor-pump which incorporated the integrated pressure sensing for motoring and pumping modes, and for varying pumping displacement. A description of the function of the control is covered in the section pertaining to the Tilt Table Simulator.

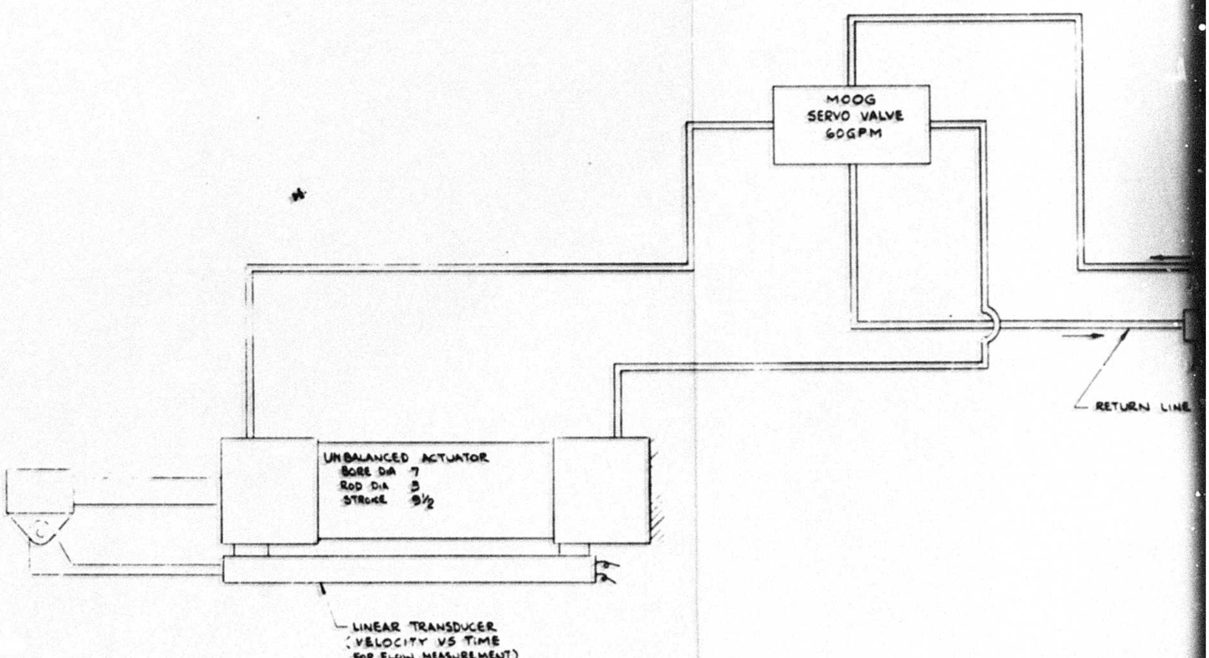
Fabrication of the Simulator consisted essentially of conversion of the Tilt Table Simulator into a test arrangement with a fixed (nontilting) mounting and the rearrangment and addition of the equipment to impose the appropriate flight control duty cycle power extraction on the energy storage substation.

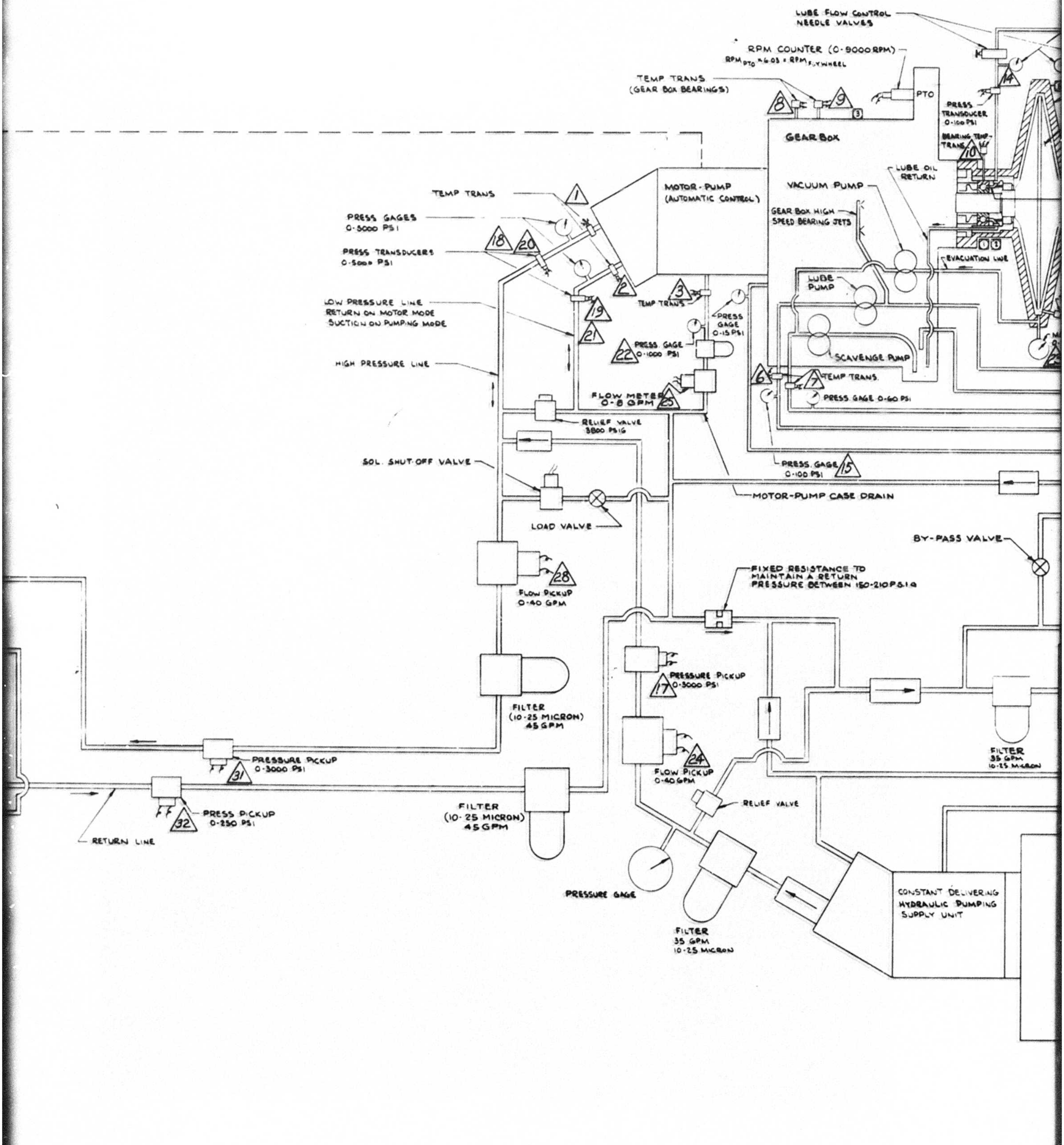
The flight control duty cycle flow demand was plotted as a curve on a rotating (Data-Trak) drum, which had a curve-following device that generated a representative signal. This signal was supplied to a Moog servo valve which imposed the flow demands on the energy storage substation by operating the actuator.

The duty cycle created for this 1,500-hour endurance test is shown in figure 44. It describes flow characteristics for a simplified yet representative set of motions for operation of three elevon panels in the XB-70 flight control system. The flow pattern shown (covering an interval of



MOTOR-PUMP CONTROL SYSTEM SCHEMATIC (AUTOMATIC CONTROL)




LUBE FLOW CONTROL NEEDLE VALVES
RPM COUNTER (0-9000 RPM)
TEMP TRANS (GEAR BOX BEARINGS)
PTO
PRESS TRANSDUCER 0-100 PSI
BEARING TEMP. TRANS.
GEAR BOX
MOTOR-PUMP (AUTOMATIC CONTROL)
LUBE OIL RETURN
VACUUM PUMP
TEMP TRANS
PRESS GAGES 0-3000 PSI
GEAR BOX HIGH SPEED BEARING JETS
PRESS TRANSDUCERS 0-5000 PSI
EVACUATION LINE
LUBE PUMP
TEMP TRANS
LOW PRESSURE LINE RETURN ON MOTOR MODE SUCTION ON PUMPING MODE
PRESS GAGE 0-15 PSI
PRESS GAGE 0-1000 PSI
SCAVENGE PUMP
HIGH PRESSURE LINE
TEMP TRANS.
PRESS GAGE 0-60 PSI
FLOW METER 0-8 GPM
RELIEF VALVE 3800 PSIG
SOL. SHUT-OFF VALVE
PRESS. GAGE 0-100 PSI
MOTOR-PUMP CASE DRAIN
LOAD VALVE
BY-PASS VALVE
FLOW PICKUP 0-40 GPM
FIXED RESISTANCE TO MAINTAIN A RETURN PRESSURE BETWEEN 150-210 PSIG
PRESSURE PICKUP 0-3000 PSI
FILTER (10-25 MICRON) 45 GPM
FILTER 35 GPM 10-25 MICRON
PRESSURE PICKUP 0-3000 PSI
FLOW PICKUP 0-40 GPM
PRESS PICKUP 0-250 PSI
RELIEF VALVE
FILTER (10-25 MICRON) 45 GPM
RETURN LINE
CONSTANT DELIVERING HYDRAULIC PUMPING SUPPLY UNIT
PRESSURE GAGE
FILTER 35 GPM 10-25 MICRON

B

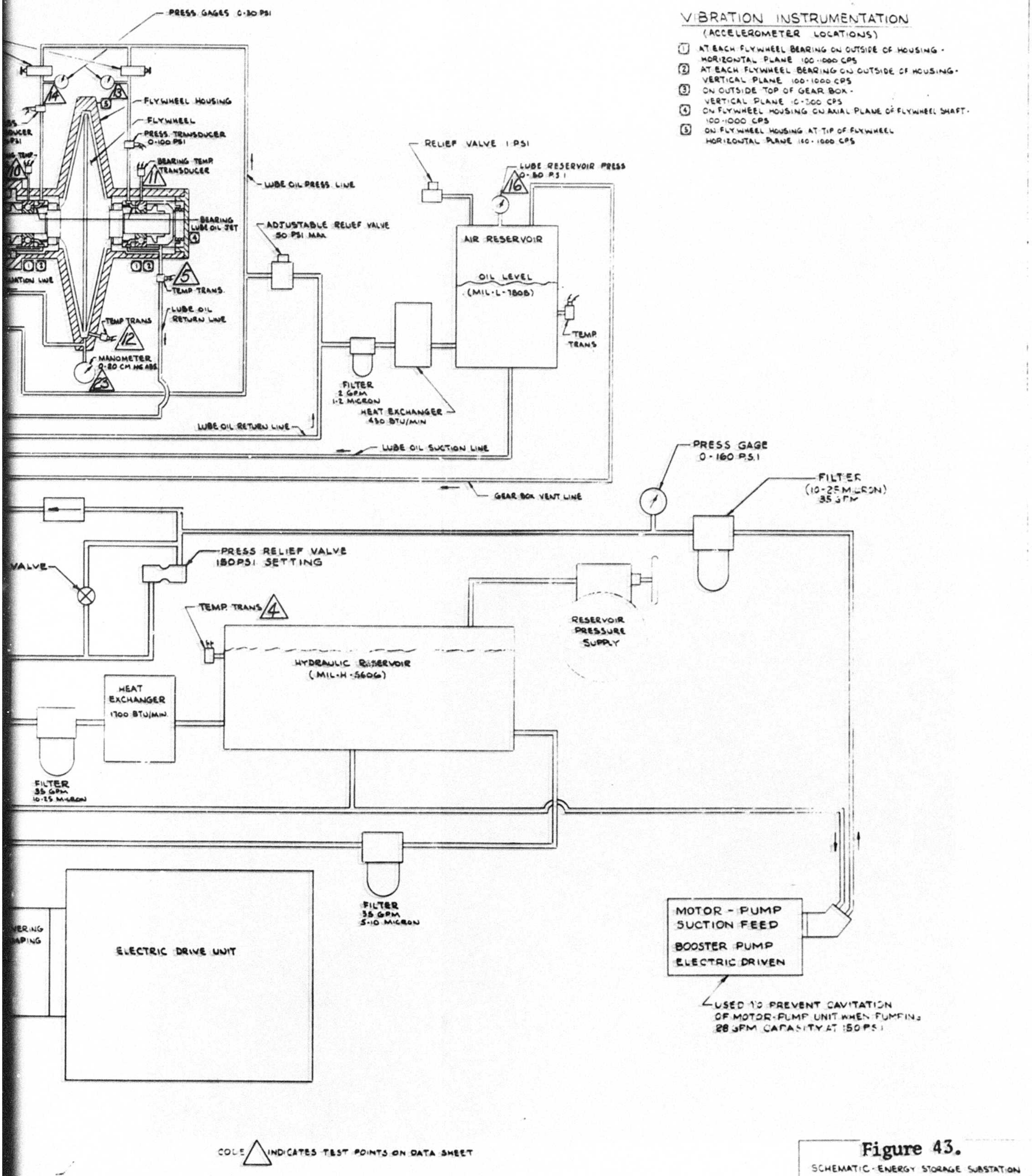
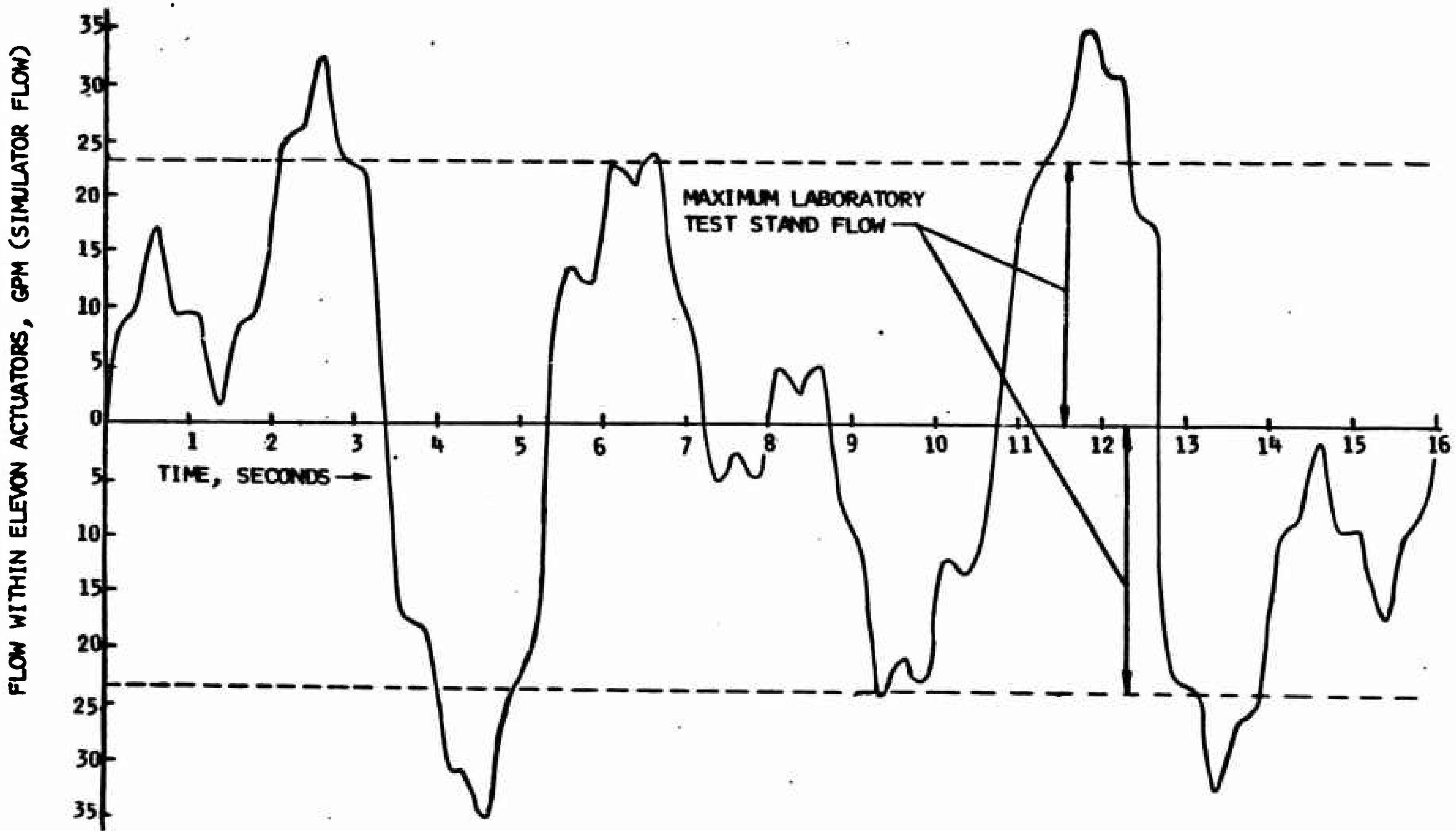


Figure 43.

SCHEMATIC - ENERGY STORAGE SUBSTATION CONTINUOUS DUTY CYCLE SIMULATOR

C



FLOW CHARACTERISTICS OF CONTINUOUS DUTY CYCLE SIMULATOR ENDURANCE TEST SPECIMEN

Figure 44. Continuous Duty Cycle Simulator

16 seconds) is considered to be one flow demand. Table XVI describes the distribution of these representative motions throughout the 5-hour flight missions which the endurance test was to demonstrate. The complete test spectrum consisted of 300 5-hour missions which were combined to demonstrate a total flight time and test time of 1,500 hours.

The first step in arriving at the above test spectrum was to establish a generalized estimate of the requirements for the flight control systems of a variety of aircraft including the XB-70 class vehicles. The mission was considered to be composed of a set of flight regimes, each of which had four types of flow demands. Table XVII is a breakdown of this information.

The duty cycle which was developed used table XVII as a demand-frequency guide. It also utilized flight-test data, analog data, and other experimental information to define the specific nature of the flow demands. The flywheel speed recovery time from these flow demands was small enough that equal spacing of the flow demands in each flight regime was acceptable and statistically compatible with pilot stick inputs. As an example the most severe flow demands occurred during emergency and attack. In this case 34 demands of 16 seconds each was accomplished within 900 seconds of elapsed time.

Surface dither (motion of the surface of small amplitude, but high frequency) was not considered to impose a load on the energy storage substation if the resultant flow demand was less than 50 percent of the maximum flow demand. These type demands would be supplied by the engine-driven pumping system in an actual air vehicle. This effectively eliminated columns A and B shown in table XVII from further consideration in constructing the test spectrum.

In the development of the test spectrum, major consideration was given to the magnitude of flow demands and the relative rates of their occurrences. The particular shape of figure 44 is a flow time history equivalent to three movements of the control surface from neutral to some deflection and then a return to neutral under varying flow demands. The three actuator movements are derived from a combination of control cycles which prevalently appear in flight-test data. The combination of flow rates was scaled to result in a 100 percent flow rate of the simulator.

C. ENDURANCE TEST

The endurance test, targeted for 1,500 hours and consisting of 300 simulated 5-hour flight missions was successfuly completed. The only incidents which arose occurred near the beginning of the test. These are discussed at greater length further on.

Figures 45 and 46 are sample data records showing the velocity of the actuator, flywheel speed, and the pressure fluctuations which appeared in the system supply. Figure 47 is similar to figures 45 and 46 except that it was made using expanded ordinate scales to clarify the action which occurs during the transition between motoring and pumping. Since actuator velocity should be proportional to actuator flow demand, a comparison between the output flow shown on these figures and the input of figure 44 was an excellent

Table XVI

BASIC DUTY CYCLE APPLICATIONS

FOR

ENDURANCE TESTING OF ENERGY STORAGE SUBSTATIONS

(APPLICABLE TO B-70 CLASS AIRCRAFT)

Mission Regime	Duration (Seconds)	Demands Per Regime C + D From Table II	Basic Duty Cycle Repetitions
Takeoff, roll	55	6	1
Lift-off, climb	1,300	8	2
Cruise	7,020	18	3
Emergency, attack	900	207	34
Cruise	7,020	18	3
Descent	1,340	24	4
Approach and rollout	380	7	1
	18,015	288	48

Table XVII

GENERALIZED CONTROL SYSTEM DUTY CYCLES

(INCLUDING B-70 CLASS AIRCRAFT)

Mission Regime	Duration (Seconds)	Surface Rate Classification * (No. of Demands Per Regime)			
		A	B	C	D
Takeoff, roll	55	55	8	4	2
Lift-off	300	300	12	2	0
Climb	1,000	1,000	24	6	0
Cruise	7,020	7,020	430	16	2
Emergency and attack	900	900	380	200	7
Cruise	7,020	7,020	430	16	2
Descent	1,340	1,340	185	24	0
Approach	300	300	98	4	0
Rollout	80	80	12	2	1

* Surface rate classifications

Class	Percent Maximum Surface Rate
A	10
B	33
C	73
D	86

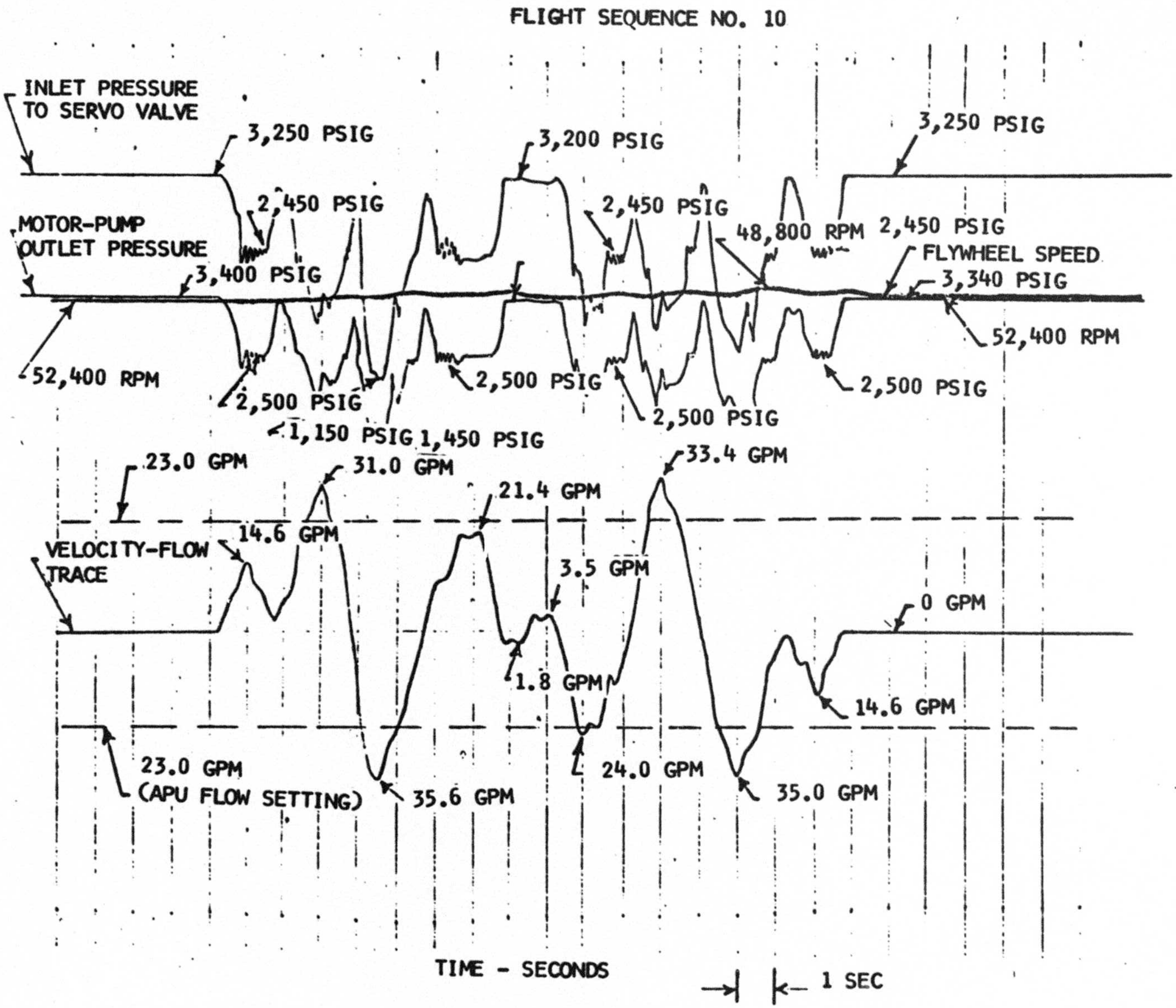


Figure 45. Continuous Duty Cycle Test Oscillograph Record

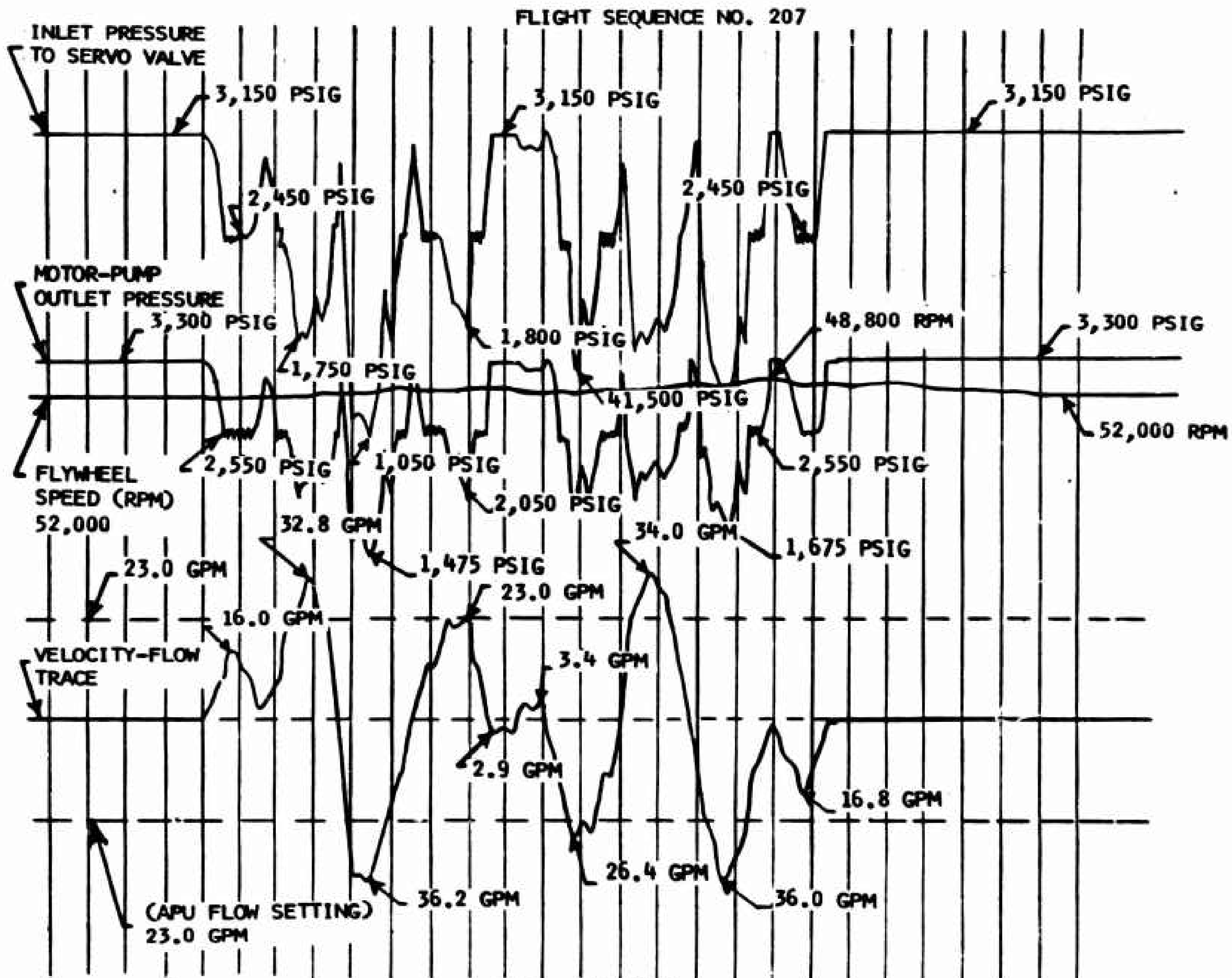


Figure 46. Continuous Duty Cycle Test Oscillograph Record

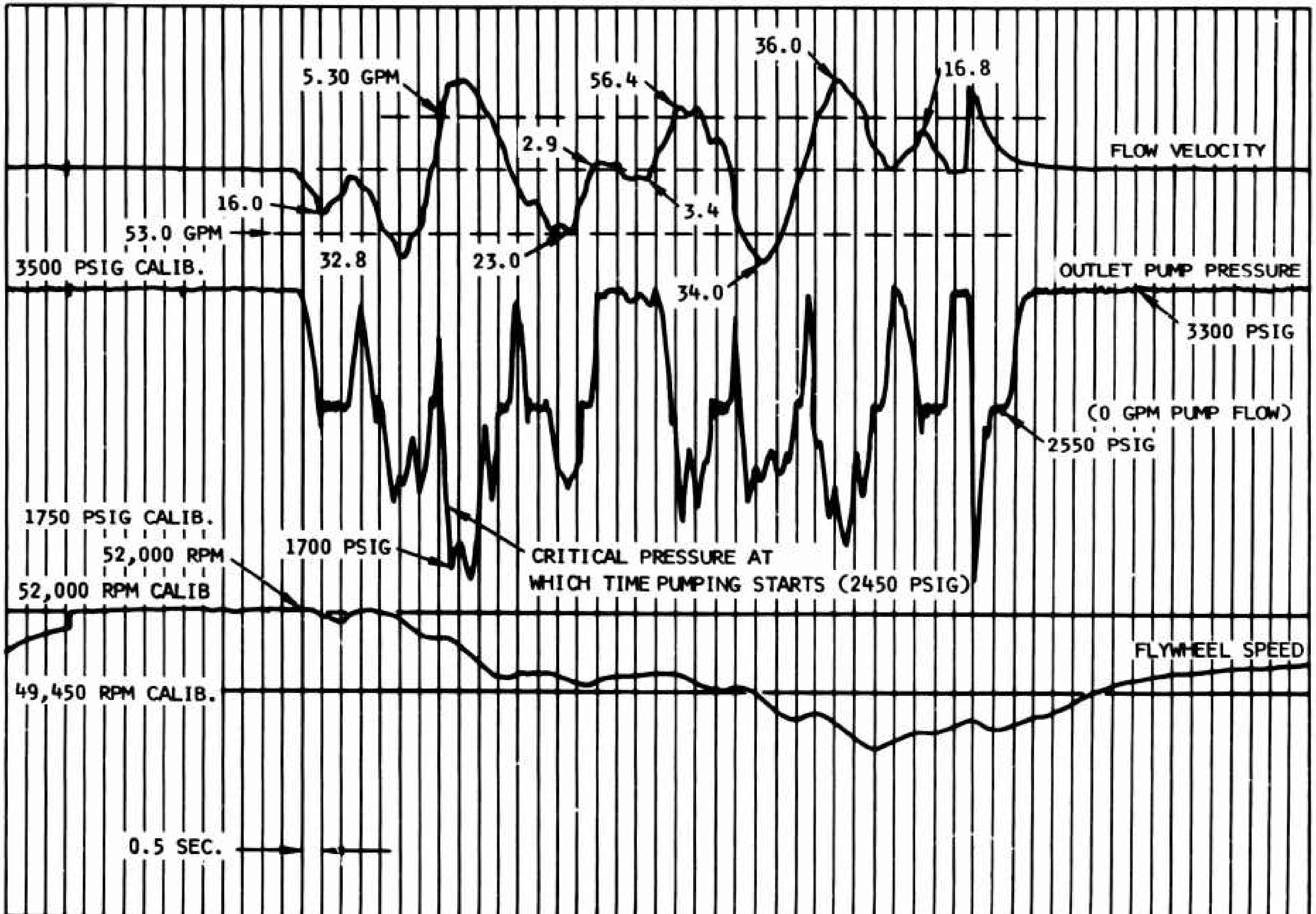


Figure 47. Oscillograph Record With Expanded Ordinate Scale

means to establish that the energy storage substation was being subjected to a representative duty cycle and was meeting its performance requirements. This was periodically done throughout the endurance test and in all cases the correlation between input and output was very good.

Table XVIII (eight pages) are data sheets taken during the tests. Each sheet was selected from among the 38 taken during the 1,500-hour test. Their selection was based upon some particular "out of the ordinary" event. These events ranged from start of test, through substation replacement due to failure to lubricant oil drain and change. Each column of data was taken at 5-hour intervals and does not show short term trends, such as a runaway temperature rise preceding bearing failure. However, they do show the long term changes, such as the gradual degradation in flywheel housing vacuum, which is discussed subsequently.

Figures 48 and 49 are plotted data points obtained from a digital computer which summarized the first 1,100 hours of testing with respect to average temperature, lubrication flow, and flywheel vacuum. The temperature plot (figure 48) is the average of the 12 temperatures taken for each of the 5-hour missions. It shows that, although variations in the operating temperature of the substation occurred, the performance was stable within a reasonable range. It should be remembered when examining the temperature range exhibited in figure 48 that the temperatures in the "shatter shield" box which surrounded the energy storage substation varied widely.

The "shatter shield" box was a device which was required by plant safety personnel until the structural failure characteristics of the flywheel were more accurately delineated. Unfortunately due to its thick woodlined walls it was an excellent insulator and prevented normal convective cooling of the substation. In an attempt to keep the temperature in the box at a normal 70° F a jury-rigged CO_2 system was used. With this arrangement, box temperatures varied widely (20° to 110° F) and helped account for the width of the temperature range in figure 48. However, the average temperature maintained in the box was approximately 70° F.

Figure 49 is a plot of the flywheel housing vacuum and is shown to be directly affected by the condition of the oil. As is noted later, the sharp dips to a lower absolute pressure are associated with routine oil changes.

During flight number 26 (125.3 hours), the flywheel vacuum gradually degraded and at a certain point the temperature of the housing suddenly climbed in a manner typical of a runaway overheat condition. At the same time it was noted that flywheel speed reduced as would be expected if significant power losses were to be suddenly introduced into the system. For these reasons the energy storage substation was shut down for investigation. There was evidence of oil leakage through the rotary seals at the flywheel hubs. However, a vacuum could still be maintained on the housing statically during bench tests, and the bearings appeared to be in operating condition. On the other hand a patch test of the lubricating system showed metal particles plus an unusual finely dispersed metallic-appearing residue. Because this type of contaminant had never been seen during previous flywheel failures it was decided to check the gearbox first. On this basis the gearbox was returned to the supplier for inspection and refurbishment.

Table XVIII

CONTINUOUS DUTY CYCLE TEST DATA

SUMMARY DATA SHEET NO: 1 DATE: 6-28-67

Running Time of Test - Hours		5.3	10.5	16.0	21.5	26.5	29.8	35.5	40.5
Vibration Amp. at Flywheel	Inboard	--	--	--	--	--	--	--	--
Bearings - g's	Outboard	--	--	--	--	--	--	--	--
(1) Flywheel Speed at Full Motoring - RPM		52,400	52.200	52,000	52,200	52,100	52,000	52,000	52,000
(2) Minimum Speed Attain Under Load - RPM		48,000	48,200	48,000	48,000	48,000	48,000	48,000	48,000
Total No. of Simulated Aircraft Flights		1	2	3	4	5	6	7	8
Total No. of Basic Duty Cycle Repetitions		48	96	144	192	240	288	336	384

NO.	TEMPERATURE DATA - DEG. F								
1	Test Pump Inlet	139	138	140	141	142	177	150	150
2	Test Pump Return	139	140	141	142	142	138	162	152
3	Test Pump Case Drain	185	182	185	182	182	188	190	196
4	Main Hyd. Reservoir Supply	130	131	135	132	134	138	132	141
5	Lube Oil Return	192	190	182	174	169	163	189	177
6	Lube Oil Suction	94	92	91	91	94	97	111	103
7	Scavange Oil Return	148	142	145	140	138	136	143	150
8	Gear Box Bearing Adjacent Pump	198	190	193	182	170	183	192	190
9	Gear Box Bearing Adjacent Flywheel	200	200	195	189	182	186	195	192
10	Inboard Flywheel Bearing	192	190	189	175	183	174	197	191
11	Outboard Flywheel Bearing	189	185	175	175	164	143	175	152
12	Flywheel Case Housing	168	155	110	104	90	120	128	100
NO.	SYSTEM PRESSURE DATA - PSIG								
13	Outboard Flywheel Bearing	30	30	30	31	30	30	31	30
14	Inboard Flywheel Bearing	33	32	32	32	30	31	32	32
15	Lube Pump Pressure Port	51	50	50	51	50	51	50	50
16	Lube Reservoir	0	0	0	0	0	0	0	0
17	Hyd Supply Pump Press. Port	3350	3350	3350	3450	3500	3440	3450	3450
18	Test Motor Press. Port (Motoring)	3350	3350	3350	3450	3500	3400	3400	3400
19	Test Motor Return Port (Motoring)	120	120	120	120	120	120	120	120
20	Test Pump Press. Range (Pumping)	2550 1450	←					→	[illegible] 1450
21	Test Pump Suction Press. (Pumping)	210 120	210 120	210 120	210 120	210 120	210 120	210 120	210 120
22	Test Motor - Pump Case Drain	210 120	210 120	210 120	210 120	210 120	210 120	210 120	210 120
23	Pressure at Flywheel Tip -- CM Hg Ab.	3.4	3.5	3.4	3.2	3.1	3.2	3.3	3.2
NO.	FLUID FLOW DATA - GPM								
24	Inlet Flow to Meter	16.0	16.0	16.0	16.0	16.0	15.8	16.0	16.0
25	Motor-Pump Case Drain Flow	0.65	0.65	0.70	0.70	0.71	0.73	0.71	0.72
26	A.P.U. Flow Rate	23.0	23.0	23.0	23.0	23.0	23.0	23.0	23.0
27	Combined A.P.U. & Test Pump Flow (Max)	36.0	36.0	36.0	35.9	35.9	35.9	35.8	35.8
28	Test Pump Flow Rate - Range	0-13	0-13	0-13	0-12.9	0-12.9	0-12.8	0-12.8	0-12.8
29	Outboard Bearing Lube Flow - cc/min	750	750	750	760	750	750	760	760
30	Inboard Bearing Lube Flow - cc/min	820	820	820	820	800	810	820	820

Table XVIII (Continued)

CONTINUOUS DUTY CYCLE TEST DATA

SUMMARY DATA SHEET NO: 2 DATE: 7-5-67

NO.	Item								
	Running Time of Test - Hours	45.5	50.5	55.5	60.5	65.5	70.5	75.8	81.0
	Vibration Amp. at Flywheel Bearings - g's — Inboard	--	--	--	--	--	--	3*	10
	Vibration Amp. at Flywheel Bearings - g's — Outboard	--	--	--	--	--	--	8*	7
	(1) Flywheel Speed at Full Motoring - RPM	52,000	51,800	51,800	51,900	52,000	51,800	51,800	51,800
	(2) Minimum Speed Attain Under Load - RPM	48,800	48,800	48,000	48,000	48,800	48,800	49,200	48,900
	Total No. of Simulated Aircraft Flights	9	10	11	12	13	14	15	16
	Total No. of Basic Duty Cycle Repetitions	432	480	528	576	624	672	720	768
NO.	TEMPERATURE DATA - DEG. F								
1	Test Pump Inlet	155	142	160	162	136	160	160	157
2	Test Pump Return	155	144	160	161	139	160	160	157
3	Test Pump Case Drain	190	190	195	195	183	193	180	190
4	Main Hyd. Reservoir Supply	133	136	145	142	130	142	155	140
5	Lube Oil Return	158	178	195	186	170	190	158	160
6	Lube Oil Suction	98	97	102	101	80	98	85	100
7	Scavange Oil Return	140	145	152	150	135	146	135	137
8	Gear Box Bearing Adjacent Pump	183	186	195	195	175	180	155	170
9	Gear Box Bearing Adjacent Flywheel	183	187	195	195	175	180	160	172
10	Inboard Flywheel Bearing	183	180	195	183	170	185	160	172
11	Outboard Flywheel Bearing	165	173	182	178	160	177	155	160
12	Flywheel Case Housing	98	100	125	108	125	120	90	105
NO.	SYSTEM PRESSURE DATA - PSIG								
13	Outboard Flywheel Bearing	31	30	29	30	30	32	30	31
14	Inboard Flywheel Bearing	30	31	30	31	30	32	30	31
15	Lube Pump Pressure Port	51	51	50	49.5	51	52	52	52
16	Lube Reservoir	0	0	0	0	0	0	0	0
17	Hyd Supply Pump Press. Port	3450	3425	3400	3400	3400	3400	3400	3400
18	Test Motor Press. Port (Motoring)	3400	3400	3375	3375	3375	3375	3375	3375
19	Test Motor Return Port (Motoring)	120	120	120	120	120	120	120	120
20	Test Pump Press. Range (Pumping)	2500 1450	←					→	2500 1450
21	Test Pump Suction Press. (Pumping)	210 120	210 120	210 120	210 120	210 120	210 120	210 120	210 120
22	Test Motor - Pump Case Drain	210 120	210 120	210 120	210 120	210 120	210 120	210 120	210 120
23	Pressure at Flywheel Tip -- CM Hg Ab.	3.2	3.2	3.2	3.1	2.4	2.9	2.6	2.7
NO.	FLUID FLOW DATA - GPM								
24	Inlet Flow to Meter	16.0	16.0	16.0	16.0	16.0	16.0	16.0	16.0
25	Motor-Pump Case Drain Flow	0.75	0.75	0.75	0.80	0.75	0.75	0.75	0.75
26	A.P.U. Flow Rate	23	23	23	23	23	23	23	23
27	Combined A.P.U. & Test Pump Flow (Max)	35.8	35.6	35.6	35.6	35.8	35.8	35.8	35.8
28	Test Pump Flow Rate - Range	0-12.8	0-12.8	0-12.8	0-12.8	0-12.8	0-12.8	0-12.8	0-12.8
29	Outboard Bearing Lube Flow - cc/min	760	750	740	740	740	760	740	760
30	Inboard Bearing Lube Flow - cc/min	820	820	820	820	810	830	810	820

*Started to record vibration g' load on inboard and outboard flywheel bearings.

Table XVIII (Continued)

CONTINUOUS DUTY CYCLE TEST DATA

SUMMARY DATA SHEET NO: 4 * DATE: 7-13-67

NO.										
	Running Time of Test - Hours		128.4	133.4	138.7	143.7	148.7	153.9	158.9	163.4
	Vibration Amp. at Flywheel Bearings - g's	Inboard	8.0	8.5	10.0	10.0	10.0	10.6	10.0	10.5
		Outboard	6.5	5.9	7.8	6.7	7.8	7.8	7.8	7.8
	(1) Flywheel Speed at Full Motoring - RPM		50,400	50,400	52,200	51,200	51,600	52,100	51,600	52,000
	(2) Minimum Speed Attain Under Load - RPM		48,000	48,000	48,000	48,000	48,000	48,000	48,000	48,000
	Total No. of Simulated Aircraft Flights		25	26	27	28	29	30	31	32
	Total No. of Basic Duty Cycle Repetitions		1200	1248	1296	1344	1392	1440	1488	1536
NO.	TEMPERATURE DATA - DEG. F									
1	Test Pump Inlet		142	149	133	162	158	100	110	101
2	Test Pump Return		142	149	150	161	158	110	111	103
3	Test Pump Case Drain		190	185	190	187	189	161	162	162
4	Main Hyd. Reservoir Supply		135	132	120	145	141	98	92	98
5	Lube Oil Return		142	138	142	145	142	160	158	166
6	Lube Oil Suction		79	73	80	82	75	88	80	93
7	Scavange Oil Return		105	111	110	120	112	111	107	115
8	Gear Box Bearing Adjacent Pump		172	170	170	183	176	170	171	178
9	Gear Box Bearing Adjacent Flywheel		148	148	155	154	150	130	158	155
10	Inboard Flywheel Bearing		163	159	145	162	145	141	148	168
11	Outboard Flywheel Bearing		142	135	135	132	140	140	141	162
12	Flywheel Case Housing		130	118	100	112	122	98	98	145
NO.	SYSTEM PRESSURE DATA - PSIG									
13	Outboard Flywheel Bearing		30	30	30	32	30	27.5	29	29
14	Inboard Flywheel Bearing		30	30	29	31.	29	27	28	28
15	Lube Pump Pressure Port		57	59	56	56	56	53	53	52
16	Lube Reservoir		0	0	0	0	0	0	0	0
17	Hyd Supply Pump Press. Port		3475	3500	3450	3450	3450	3400	3400	3400
18	Test Motor Press. Port (Motoring)		3460	3450	3400	3400	3400	3350	3350	3375
19	Test Motor Return Port (Motoring)		120	120	120	120	120	135	135	110
20	Test Pump Press. Range (Pumping)		2550/1450	←					→	2550/1450
21	Test Pump Suction Press. (Pumping)		210/120	210/120	210/120	210/120	210/120	275/135	275/135	200/110
22	Test Motor - Pump Case Drain		210/120	210/120	210/120	210/120	210/120	275/135	275/135	200/110
23	Pressure at Flywheel Tip -- CM Hg Ab.		4.8	3.8	3.4	4.8	2.9	2.7	2.7	2.0
NO.	FLUID FLOW DATA - GPM									
24	Inlet Flow to Meter		15.5	15.5	16.0	16.0	16.0	16.0	16.0	16.0
25	Motor-Pump Case Drain Flow		0.65	0.65	0.75	0.75	0.75	0.60	0.60	0.62
26	A.P.U. Flow Rate		23	23	23	23	23	23	23	23
27	Combined A.P.U. & Test Pump Flow (Max)		36.0	36.1	36.1	36.1	36.0	36.0	36.0	36.0
28	Test Pump Flow Rate - Range		0-13	0-13.1	0-13.1	0-13.1	0-13	0-13	0-13	0-13
29	Outboard Bearing Lube Flow - cc/min		465	465	465	475	465	425	450	450
30	Inboard Bearing Lube Flow - cc/min		485	485	465	470	465	445	455	455

*Testing of flywheel sub-station No. 2 began at 125.3 hr. into the program.

Table XVIII (Continued)

CONTINUOUS DUTY CYCLE TEST DATA

SUMMARY DATA SHEET NO: 5 * DATE: 7-19-67

Running Time of Test - Hours			168.9	174.2	179.4	184.4	189.4	194.4	199.4	203.8
Vibration Amp. at Flywheel Bearings - g's		Inboard	13.6	10.7	9.7	10.3	12.0	11.6	11.5	11.0
		Outboard	8.5	7.8	7.8	7.8	7.8	7.8	7.8	6.5
(1) Flywheel Speed at Full Motoring - RPM			52,000	52,000	52,000	52,000	52,000	52,000	52,000	52,000
(2) Minimum Speed Attain Under Load - RPM			48,000	48,000	48,000	48,000	48,000	48,000	48,000	48,500
Total No. of Simulated Aircraft Flights			33	34	35	36	37	38	39	40
Total No. of Basic Duty Cycle Repetitions			1584	1632	1680	1728	1776	1824	1872	1920
NO.	TEMPERATURE DATA - DEG. F									
1	Test Pump Inlet		102	110	115	112	100	101	122	103
2	Test Pump Return		105	110	115	113	100	102	121	103
3	Test Pump Case Drain		160	160	155	157	155	156	132	160
4	Main Hyd. Reservoir Supply		92	92	90	91	90	94	114	100
5	Lube Oil Return		158	165	165	165	170	157	138	153
6	Lube Oil Suction		85	90	90	89	95	95	111	100
7	Scavange Oil Return		105	130	122	125	140	118	110	125
8	Gear Box Bearing Adjacent Pump		168	180	168	168	190	169	147	175
9	Gear Box Bearing Adjacent Flywheel		154	180	168	168	190	169	150	170
10	Inboard Flywheel Bearing		150	165	140	140	170	151	140	150
11	Outboard Flywheel Bearing		145	155	140	140	170	145	141	150
12	Flywheel Case Housing		106	140	130	132	165	129	130	135
NO.	SYSTEM PRESSURE DATA - PSIG									
13	Outboard Flywheel Bearing		30	32	29	29	34	32	32	32
14	Inboard Flywheel Bearing		30	32	30	30	32	30	30	31
15	Lube Pump Pressure Port		56	57	56	56	57	56	56	55
16	Lube Reservoir		0	0	0	0	0	0	0	0
17	Hyd Supply Pump Press. Port		3400	3400	3400	3400	3400	3400	3400	3400
18	Test Motor Press. Port (Motoring)		3375	3375	3380	3375	3375	3375	3375	3375
19	Test Motor Return Port (Motoring)		120	110	110	110	120	120	112	110
20	Test Pump Press. Range (Pumping)		2550 1450							2550 1450
21	Test Pump Suction Press. (Pumping)		210 120	210 110	210 110	210 110	210 120	210 120	210 112	210 110
22	Test Motor - Pump Case Drain		210 120	210 110	210 110	210 110	210 120	210 120	210 112	210 110
23	Pressure at Flywheel Tip -- CM Hg Ab.		2.4	2.5	2.2	2.3	2.7	2.2	2.6	2.6
NO.	FLUID FLOW DATA - GPM									
24	Inlet Flow to Meter		16.0	16.0	16.0	16.0	16.0	16.0	16.0	16.0
25	Motor-Pump Case Drain Flow		0.60	0.60	0.65	0.65	0.65	0.65	0.65	0.65
26	A.P.U. Flow Rate		23	23	23	23	23	23	23	23
27	Combined A.P.U. & Test Pump Flow (Max)		36.0	35.9	35.9	35.8	35.9	35.9	36.0	36.0
28	Test Pump Flow Rate - Range		0-13	0-12.9	0-12.9	0-12.8	0-12.8	0-12.9	0-13	0-13
29	Outboard Bearing Lube Flow - cc/min		465	760	750	750	800	740	740	740
30	Inboard Bearing Lube Flow - cc/min		485	830	800	800	820	790	790	805

* 2nd start of flywheel sub-station No. 1 after replacement of bearings and seals.

Table XVIII (Continued)

CONTINUOUS DUTY CYCLE TEST DATA

SUMMARY DATA SHEET NO: 10   *   DATE: 8-8-67

Running Time of Test - Hours		368.5	373.7	378.8	383.9	388.7	394.3	399.0	404.0
Vibration Amp. at Flywheel	Inboard	10.8	10.2	7.8	10.3	10.8	7.4	10.9	10.3
Bearings - g's	Outboard	6.5	7.3	6.1	6.5	7.8	6.5	6.7	7.8
(1) Flywheel Speed at Full Motoring - RPM		52000	52000	52400	52400	52200	52200	52200	52200
(2) Minimum Speed Attain Under Load - RPM		49000	49000	49000	49000	49000	49000	49000	49000
Total No. of Simulated Aircraft Flights		73	74	75	76	77	78	79	80
Total No. of Basic Duty Cycle Repetitions		3504	3552	3600	3648	3696	3744	3792	3840

NO.	TEMPERATURE DATA - DEG. F								
1	Test Pump Inlet	93	93	98	90	95	95	95	98
2	Test Pump Return	102	102	102	100	100	98	100	105
3	Test Pump Case Drain	145	140	138	138	140	138	140	143
4	Main Hyd. Reservoir Supply	90	90	85	85	90	85	90	95
5	Lube Oil Return	150	167	141	160	170	158	170	170
6	Lube Oil Suction	90	85	80	80	85	80	80	90
7	Scavange Oil Return	120	120	115	120	125	118	120	125
8	Gear Box Bearing Adjacent Pump	160	163	158	160	165	160	160	165
9	Gear Box Bearing Adjacent Flywheel	165	175	160	170	180	170	170	180
10	Inboard Flywheel Bearing	150	160	152	150	170	165	160	165
11	Outboard Flywheel Bearing	140	155	138	145	155	140	155	157
12	Flywheel Case Housing	120	130	105	120	155	135	140	145
NO.	SYSTEM PRESSURE DATA - PSIG								
13	Outboard Flywheel Bearing	34	34	33	33	34	33	34	33
14	Inboard Flywheel Bearing	34	34	33	33	34	33	34	33
15	Lube Pump Pressure Port	56	55	57	56	55	57	56	56
16	Lube Reservoir	0	0	0	0	0	0	0	0
17	Hyd Supply Pump Press. Port	3350	3350	3350	3350	3350	3350	3350	3350
18	Test Motor Press. Port (Motoring)	3300	3300	3300	3300	3300	3300	3300	3300
19	Test Motor Return Port (Motoring)	115	115	115	115	115	115	115	115
20	Test Pump Press. Range (Pumping)	2550 1475	←					→	2550 1475
21	Test Pump Suction Press. (Pumping)	115 210	115 210	115 210	115 210	115 210	115 210	115 210	115 210
22	Test Motor - Pump Case Drain	115 210	115 210	115 210	115 210	115 210	115 210	115 210	115 210
23	Pressure at Flywheel Tip - CM Hg Ab.	3.1	3.2	3.0	3.1	3.1	3.1	3.2	3.2
NO.	FLUID FLOW DATA - GPM								
24	Inlet Flow to Meter	16.0	16.0	16.0	16.0	16.0	16.0	16.0	16.0
25	Motor-Pump Case Drain Flow	0.65	0.70	0.65	0.65	0.70	0.67	0.67	0.70
26	A.P.U. Flow Rate	23	23	23	23	23	23	23	23
27	Combined A.P.U. & Test Pump Flow (Max)	36.0	36.0	36.0	36.0	36.0	35.9	35.9	36.0
28	Test Pump Flow Rate - Range	0-13	0-13	0-13	0-13	0-13	0-12.9	0-12.9	0-13
29	Outboard Bearing Lube Flow - cc/min	770	770	765	765	770	765	770	765
30	Inboard Bearing Lube Flow - cc/min	830	830	825	825	830	825	830	825

* NOTE: Min. speed during "attach" 49,500 RPM. Recovered to 51,500 between cycles with 52,400 max. regulated speed after "attack".

CONTINUOUS DUTY CYCLE TEST DATA

SUMMARY DATA SHEET NO: 16 * DATE: 8-29-67

NO.	Item								
	Running Time of Test - Hours	610.0	615.1	620.2	625.1	630.2	635.0	640.0	645.0
	Vibration Amp. at Flywheel Bearings - g's: Inboard	9.7	9.7	9.7	9.1	9.1	9.7	9.7	9.7
	Vibration Amp. at Flywheel Bearings - g's: Outboard	6.5	6.5	6.5	5.7	6.5	6.5	5.7	5.7
	(1) Flywheel Speed at Full Motoring - RPM	52000	52000	52000	52000	52000	52000	52000	52000
	(2) Minimum Speed Attain Under Load - RPM	49000	49000	49000	49000	49000	49000	49000	49000
	Total No. of Simulated Aircraft Flights	121	122	123	124	125	126	127	128
	Total No. of Basic Duty Cycle Repetitions	5908	5956	6004	6052	6100	6148	6196	6244
NO.	TEMPERATURE DATA - DEG. F								
1	Test Pump Inlet	180	105	85	82	88	100	90	88
2	Test Pump Return	113	115	96	93	98	102	100	98
3	Test Pump Case Drain	160	160	148	143	147	148	145	145
4	Main Hyd. Reservoir Supply	100	105	85	82	90	85	90	88
5	Lube Oil Return	160	160	152	153	152	160	160	160
6	Lube Oil Suction	100	105	90	88	90	87	90	90
7	Scavange Oil Return	135	135	125	123	128	125	130	127
8	Gear Box Bearing Adjacent Pump	175	175	172	170	170	170	170	165
9	Gear Box Bearing Adjacent Flywheel	175	175	172	170	170	170	170	165
10	Inboard Flywheel Bearing	175	170	165	163	165	168	168	160
11	Outboard Flywheel Bearing	160	145	150	148	145	150	155	153
12	Flywheel Case Housing	155	120	145	148	140	135	145	140
NO.	SYSTEM PRESSURE DATA - PSIG								
13	Outboard Flywheel Bearing	34	35	35	35	35	35	34	34
14	Inboard Flywheel Bearing	34	35	35	35	35	35	34	34
15	Lube Pump Pressure Port	54	56	57	57	57	56	56	56
16	Lube Reservoir	0	0	0	0	0	0	0	0
17	Hyd Supply Pump Press. Port	3350	3350	3350	3350	3350	3350	3350	3350
18	Test Motor Press. Port (Motoring)	3300	3300	3300	3300	3300	3300	3300	3300
19	Test Motor Return Port (Motoring)	120	120	120	120	120	120	120	125
20	Test Pump Press. Range (Pumping)	2550 1450	←					→	2550 1450
21	Test Pump Suction Press. (Pumping)	120 210	120 210	120 210	120 210	120 210	120 210	120 210	120 210
22	Test Motor - Pump Case Drain	120 210	120 210	120 210	120 210	120 210	120 210	120 210	120 210
23	Pressure at Flywheel Tip -- CM Hg Ab.	3.3	2.1	2.1	2.1	2.2	2.2	2.2	2.2
NO.	FLUID FLOW DATA - GPM								
24	Inlet Flow to Meter	16.0	16.0	16.0	16.0	16.0	16.0	16.0	16.0
25	Motor-Pump Case Drain Flow	0.70	0.70	0.70	0.65	0.70	0.69	0.69	0.65
26	A.P.U. Flow Rate	23	23	23	23	23	23	23	23
27	Combined A.P.U. & Test Pump Flow (Max)	36.0	36.0	36.1	36.0	36.0	36.0	36.1	36.1
28	Test Pump Flow Rate - Range	0-13	0-13	0-13	0-13	0-13	0-13	0-13.1	0-13.1
29	Outboard Bearing Lube Flow - cc/min	770	775	775	775	775	775	770	770
30	Inboard Bearing Lube Flow - cc/min	830	835	835	835	835	835	830	830

* Changed MIL-L-7808 Lube Oil at 614 hours of testing. This improved system vacuum from 3.3 cm. ab. to 2.1 cm. ab.

Table XVIII (Continued)

CONTINUOUS DUTY CYCLE TEST DATA

SUMMARY DATA SHEET NO: 29 DATE: 10-16-67

Running Time of Test - Hours		1139.0	1144.0	1149.0	1154.0	1159.0	1164.0	1169.0	1174.0
Vibration Amp. at Flywheel Bearings - g's	Inboard	7.3	9.7	9.1	7.9	8.5	8.9	10.1	9.1
	Outboard	3.3	-	5.8	5.5	3.7	3.9	7.3	4.5
(1) Flywheel Speed at Full Motoring - RPM		52,100	52,000	52,000	52,000	52,000	52,000	52,150	52,000
(2) Minimum Speed Attain Under Load - RPM		48,500	48,500	48,500	48,500	48,500	48,500	48,500	48,500
Total No. of Simulated Aircraft Flights		225	226	227	228	229	230	231	232
Total No. of Basic Duty Cycle Repetitions		10,800	10,848	10,896	10,944	10,992	11,040	11,088	11,136
NO.	TEMPERATURE DATA - DEG. F								
1	Test Pump Inlet	89	95	100	100	93	90	100	92
2	Test Pump Return	89	95	100	100	93	90	100	92
3	Test Pump Case Drain	148	152	142	148	148	150	145	145
4	Main Hyd. Reservoir Supply	80	88	80	82	83	83	80	78
5	Lube Oil Return	158	170	163	160	158	170	152	155
6	Lube Oil Suction	85	95	81	88	88	93	80	80
7	Scavenge Oil Return	120	132	128	130	128	133	112	122
8	Gear Box Bearing Adjacent Pump	175	188	172	180	185	191	171	180
9	Gear Box Bearing Adjacent Flywheel	170	188	172	180	185	191	171	180
10	Inboard Flywheel Bearing	122	175	161	160	159	170	152	160
11	Outboard Flywheel Bearing	150	165	152	152	150	163	159	150
12	Flywheel Case Housing	103	142	108	102	100	112	112	110
NO.	SYSTEM PRESSURE DATA - PSIG								
13	Outboard Flywheel Bearing	33	35	34	34	34	36	34	35
14	Inboard Flywheel Bearing	34	35	34	34	35	34	33	35
15	Lube Pump Pressure Port	56	57	58	57	58	58	59	59
16	Lube Reservoir	0	0	0	0	0	0	0	0
17	Hyd Supply Pump Press. Port	3350	3350	3350	3350	3350	3350	3350	3350
18	Test Motor Press. Port (Motoring)	3300	3300	3300	3300	3300	3300	3300	3300
19	Test Motor Return Port (Motoring)	115	115	115	115	115	115	115	115
20	Test Pump Press. Range (Pumping)	1450-2450	1450-2450	1450-2450	1450-2450	1450-2450	1450-2450	1450-2450	1450-2450
21	Test Pump Suction Press. (Pumping)	115-200	115-200	115-200	115-200	115-200	115-200	115-200	115-200
22	Test Motor - Pump Case Drain	115-200	115-200	115-200	115-200	115-200	115-200	115-200	115-200
23	Pressure at Flywheel Tip - CM Hg Ab.	3.0	2.0	2.1	2.1	2.2	2.3	2.2	2.3
NO.	FLUID FLOW DATA - GPM								
24	Inlet Flow to Meter	16.1	16.1	16.1	16.1	16.1	16.1	16.1	16.1
25	Motor-Pump Case Drain Flow	0.72	0.75	0.75	0.75	0.75	0.75	0.75	0.75
26	A.P.U. Flow Rate	23	23	23	23	23	23	23	23
27	Combined A.P.U. & Test Pump Flow (Max)	36.1	36.1	36.1	36.1	36.0	36.0	36.0	36.0
28	Test Pump Flow Rate - Range	0-13.1	0-13.1	0-13.1	0-13.1	0-13	0-13	0-13	0-13
29	Outboard Bearing Lube Flow - cc/min	770	790	810	810	810	815	815	790
30	Inboard Bearing Lube Flow - cc/min	860	870	870	870	870	840	870	870

*Changed MIL-L-7808 lube oil at 1139.0 hours of testing. This improved system vacuum from 3.0 cm. ab. to 2.0 cm. ab.

Table XVIII (Concluded)

CONTINUOUS DUTY CYCLE TEST DATA

SUMMARY DATA SHEET NO: 38 DATE: 11-16-67

Running Time of Test - Hours		1499.0	1504.0	1509.0	1514.1
Vibration Amp. at Flywheel Bearings - g's	Inboard	7.9	7.8	6.1	6.7
	Outboard	6.5	6.8	7.3	7.1
(1) Flywheel Speed at Full Motoring - RPM		52,000	52,000	52,000	52,000
(2) Minimum Speed Attain Under Load - RPM		48,500	48,500	48,500	48,500
Total No. of Simulated Aircraft Flights		297	298	299	300
Total No. of Basic Duty Cycle Repetitions		14256	14304	14352	14400
NO.	**TEMPERATURE DATA - DEG. F**				
1	Test Pump Inlet	94	92	95	100
2	Test Pump Return	94	90	95	100
3	Test Pump Case Drain	138	130	138	140
4	Main Hyd. Reservoir Supply	98	90	85	95
5	Lube Oil Return	155	145	142	125
6	Lube Oil Suction	80	80	85	90
7	Scavange Oil Return	120	120	128	125
8	Gear Box Bearing Adjacent Pump	160	160	172	155
9	Gear Box Bearing Adjacent Flywheel	165	165	176	170
10	Inboard Flywheel Bearing	158	160	160	160
11	Outboard Flywheel Bearing	145	140	148	150
12	Flywheel Case Housing	132	130	140	125
NO.	**SYSTEM PRESSURE DATA - PSIG**				
13	Outboard Flywheel Bearing	36	36	36	36
14	Inboard Flywheel Bearing	36	36	36	36
15	Lube Pump Pressure Port	55	55	56	55
16	Lube Reservoir	0	0	0	0
17	Hyd Supply Pump Press. Port	3350	3350	3350	3350
18	Test Motor Press. Port (Motoring)	3300	3300	3300	3300
19	Test Motor Return Port (Motoring)	125	125	125	125
20	Test Pump Press. Range (Pumping)	1450 2450	1450 2450	1450 2450	1450 2450
21	Test Pump Suction Press. (Pumping)	[illegible] 200	[illegible] 200	[illegible] 200	[illegible] 200
22	Test Motor - Pump Case Drain	125 200	125 200	125 200	125 200
23	Pressure at Flywheel Tip -- CM Hg Ab.	2.7	2.7	2.7	2.8
NO.	**FLUID FLOW DATA - GPM**				
24	Inlet Flow to Meter	16.6	16.6	16.6	16.6
25	Motor-Pump Case Drain Flow	0.75	0.70	0.75	0.75
26	A.P.U. Flow Rate	23	23	23	23
27	Combined A.P.U. & Test Pump Flow (Max)	35.9	35.9	35.8	35.9
28	Test Pump Flow Rate - Range	0-12.9	0-12.9	0-12.8	0-12.9
29	Outboard Bearing Lube Flow - cc/min	828	828	828	828
30	Inboard Bearing Lube Flow - cc/min	880	880	880	880

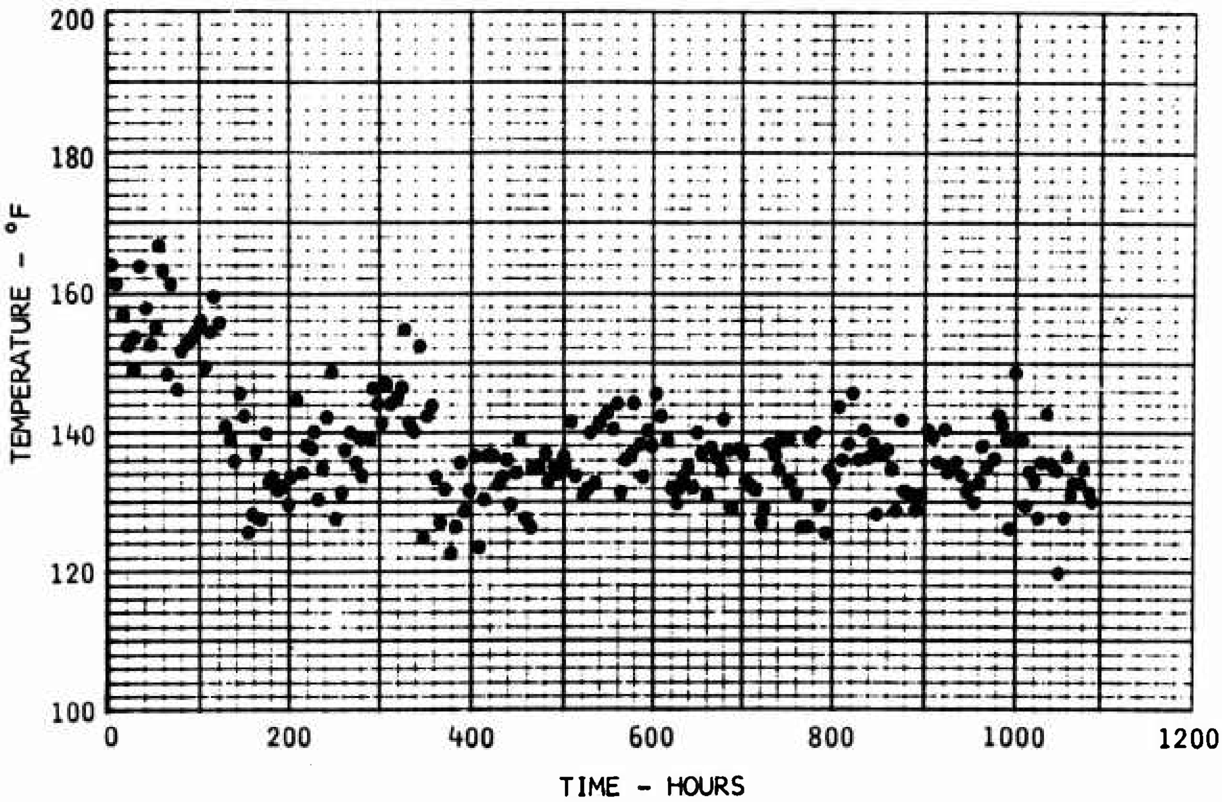


Figure 48. Average Substation Temperature

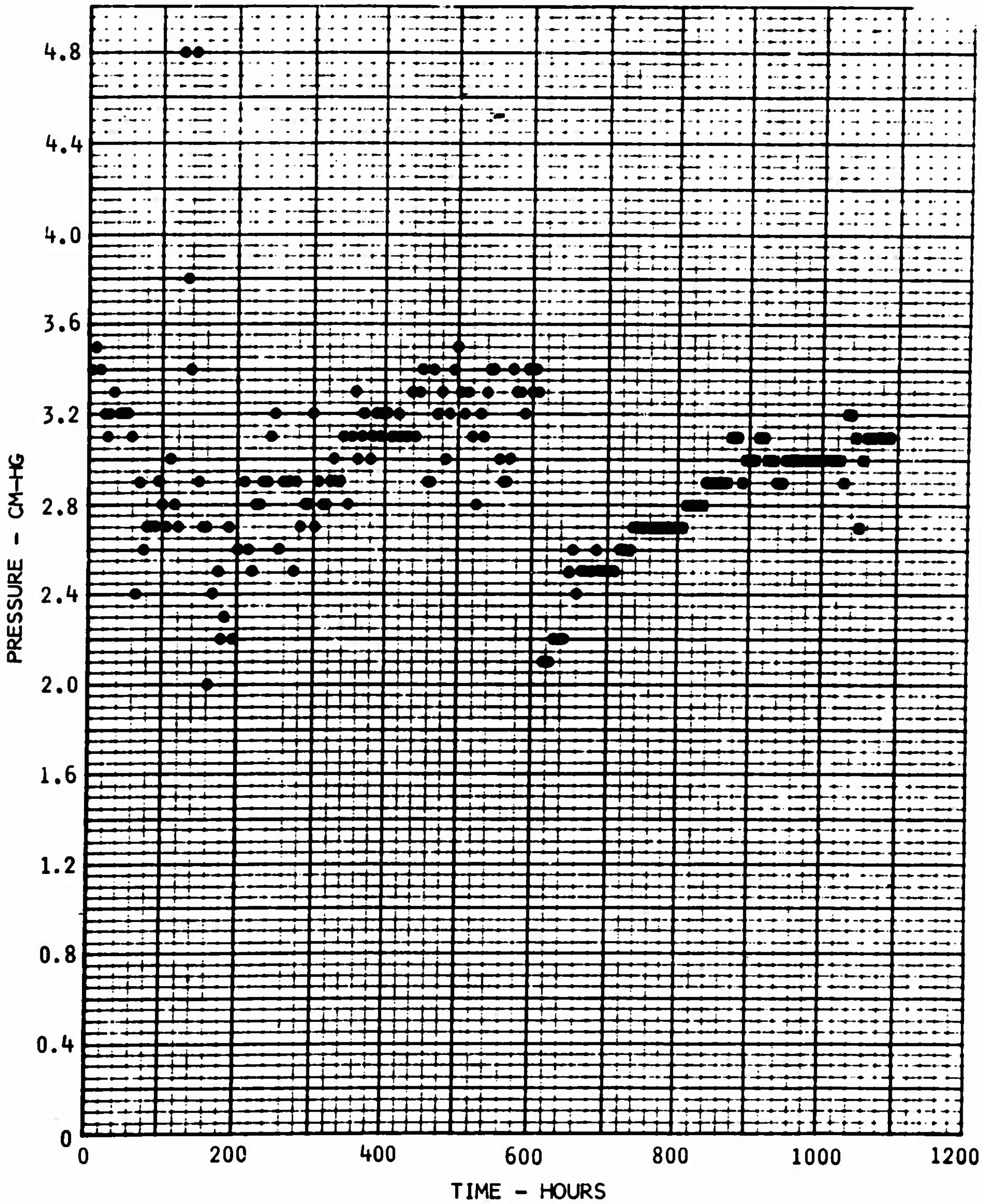


Figure 49. Flywheel Absolute Pressure

All gear tooth faces were in perfect condition as well as all of the bearings. The only parts replaced in the gearbox upon reassembly consisted of replacing the seals as a precautionary measure. There was some sign of fretting corrosion at the gearbox end of the flywheel to gearbox quill shaft. However, there was no explanation for the peculiar contaminant.

The flywheel was then disassembled for lack of other failure evidence. The source of the peculiar contaminant became readily apparent when several drops of mercury were found in the flywheel housing. It is believed that mercury came from the manometer used to measure absolute vacuum.

The carbon seal faces were worn, one more badly than the other. The mating ring which was paired with the more seriously worn seal had a radial scratch across the sealing area. The scratch probably occurred during assembly. It caused scoring of the carbon seal face and led to the early leakage failure.

The tip of the flywheel was discolored from heat and it is estimated that a temperature in excess of 600° F was reached. A surface hardness test indicated that the tensile strength dropped from 260 to 210 ksi for the outer 1/2-inch of the periphery. (This was not considered to impair the strength of the flywheel, since the outer edge has about a 20 percent lesser stress than the center of the flywheel.)

In addition, but not related to the above, a flywheel support bearing was found to have a cracked separator ring.

Microscopic examination of the patch taken from the lubricating oil later disclosed that the metallic particles were about 90 percent mercury.

A complete replacement of the lubrication oil was made as well as the washing and flushing of all tubes, components, and passages which had contact with the oil. A separate recirculating filter system was also connected to the oil reservoir, and was operated on a continuous 24-hour-a-day basis.

In the interim, while the Number 1 energy storage substation was being refurbished, the Number 2 substation was installed on the simulator. It operated satisfactorily for 48.9 hours except in one respect. The vacuum would fluctuate between two relatively fixed levels (i.e., 5.2 and 2.8 centimeters of mercury absolute). During flight number 35, a runaway temperature condition was detected at the flywheel housing, that resembled the condition noted during flight number 26 except of lesser magnitude.

Testing of the Number 2 substation was stopped and investigation of the condition revealed that failure of the rotary seals had allowed oil to enter the housing. There was no discoloration of the flywheel. It is believed that this failure and the previous one were identical except for the fact that, in the first case, mercury had been allowed to accumulate and was subsequently drawn into the housing, from the vacuum instrumentation line. It was felt that the 600° F tip temperature resulted solely from the presence of mercury with its high density and associated high drag losses and that these temperatures will not be reached in the presence of oil.

Refurbishment of this flywheel revealed that the sealing faces of both seals were in generally satisfactory condition. However, one of the dynamic seals was noted to have lost much of its ability to return the carbon face to an extended position upon being depressed slightly beyond the installed position. Two possible causes were investigated: one was an improper installation procedure wherein the spring was overstressed; the other was a degradation of the O-ring seal by swelling. Investigation revealed the probabilities were that the seal housing had been deformed axially during assembly and the seal spring overstressed. In any event the "sticky" action of the seal was believed to be the cause for the erratic performance of the vacuum system.

A change was incorporated in the vacuum instrumentation wherein the test was continuously monitored by a Bourdon-type vacuum gage. The mercury column manometer was used momentarily only once per flight to obtain an accurate reading. This was done to avoid a repetition of the mercury filling incident.

From the time of the reinstallation of energy storage substation Number 1, at 174.2 hours of total test time, the simulator operated satisfactorily. However, during one of the shutdown periods the various heat exchangers for the oil systems were cleaned and flushed. This resulted in a considerable improvement in their efficiency, and the energy storage substation operated at generally lower temperatures from that time on.

During the course of operating, it was noted that the vacuum created in the flywheel case slowly degraded. Absolute pressure readings when energy storage substation Number 1 was rebuilt and reinstalled at 174.2 hours were around 2.5 centimeters of mercury. At 614 hours of operation pressure readings had gradually increased to 3.1 centimeters of mercury.

It was discovered, upon making a routine replacement of the MIL-L-7808 lubricating oil, that the vacuum returned to its original condition. It is reasoned from these observations that the improvement was brought about by the effect of increased viscosity of the new oil and its effect upon the sealing characteristics of the vacuum pump gear teeth as well as the rotary shaft seals. The gradual degradation in vacuum level and its sharp recovery upon changing oil can be seen in table XVIII, sheets 5, 10, and 16, as well as in figure 49.

During the repeated duty cycles representing peak energy extraction, the flywheel speed was reduced only 8 percent, and it recovered to threshold regulated speed in only 3 seconds. The flywheel weighed 15 pounds; however, from the above values it is easily recognized that the optimum flywheel would have been considerably smaller and lighter with a larger (10 to 15 percent) speed reduction. Conversely, maintaining the same weight and size, this flywheel would be capable of meeting a larger percentage of the peak power demand, thus making possible further reductions in the basic power generation and distribution system. The system as tested met one third of the peak power demand. With a somewhat larger motor-pump and greater ($\approx$15 percent) speed reduction it could supply one half of the peak power demand.

The total test imposed maximum operational times on components as follows:

Flywheel	*1,576.6
Gearbox	*1,596.6
Motor-pump	1,523.6
Rotary carbon seals (52,000 rpm)	1,339.9
Flywheel support bearings	1,339.9

*Including tilt table test time and various checkout operations

D. DISASSEMBLY AND INSPECTION

Following completion of the endurance test (1,514.1 hours of operation) the energy storage substation was disassembled (see figure 50) and inspected. The condition of the components parts was found to be as follows:

FLYWHEEL HOUSING

The flywheel housing was in excellent condition. However, it contained very minute traces of mercury which was believed to have originated from the absolute pressure manometer used to obtain accurate flywheel evacuated atmospheric pressure data once during each 5-hour flight. (A bourdon tube pressure gage was used for continuous monitoring of the test.) In addition to the mercury, a small amount of black carbon paste, believed to have been worn from the rotary seals was evident near the seal on one side only. The flywheel cavity was otherwise clean and free of all foreign material. It would appear that the minute oil leakage past the carbon seal faces had a washing action and had carried residue into the vacuum pump. A photograph of the disassembled flywheel housing is shown in figure 51.

FLYWHEEL

The flywheel was blackened with oxidized lubricant residue. This coating was sufficiently tough that removal by abrasive rubbing was required to inspect the parent surface of the metal. The metal surface underneath had its original heat treatment oxide coating except for the outer 1-1/2 inches of periphery. The outermost 1/2 inch was blue and the next inward 1/2 inch was purple. From 1 inch inward the color faded from dark straw to the original color when manufactured.

A hardness check indicated that the tensile strength was unchanged since the mercury filling episode which occurred at 125.3 hours. Figure 52 shows the flywheel and the abrasively-cleaned strip.

The before and after measurements of the outer diameter were not recorded for comparison; however, from the fit of the wheel in the housing, no detectable change was observed. It is believed that no growth had occurred, since the balance of the wheel had not displayed any change as of the end of the endurance test.

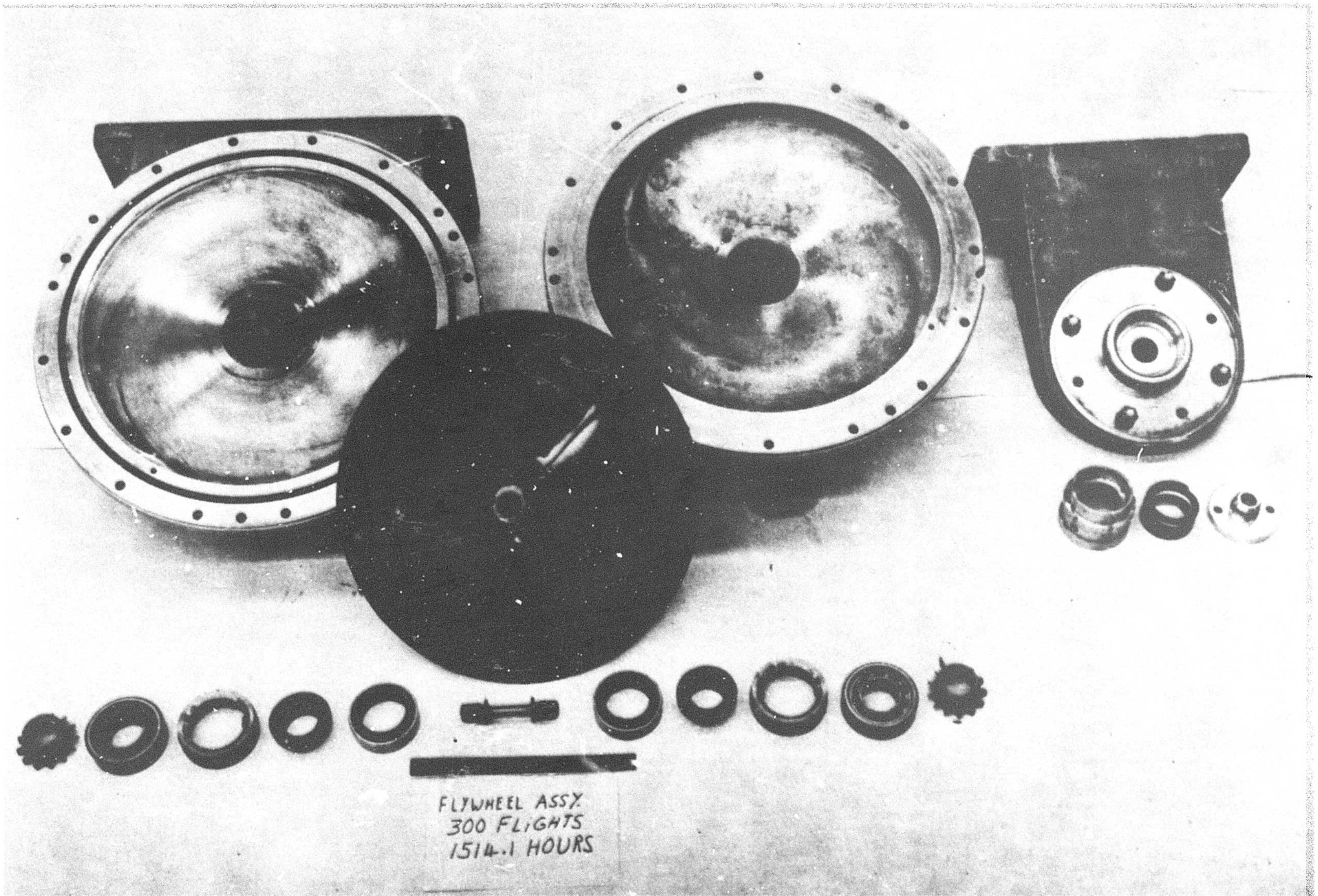


Figure 50. Continuous Duty Endurance Test Flywheel Assembly

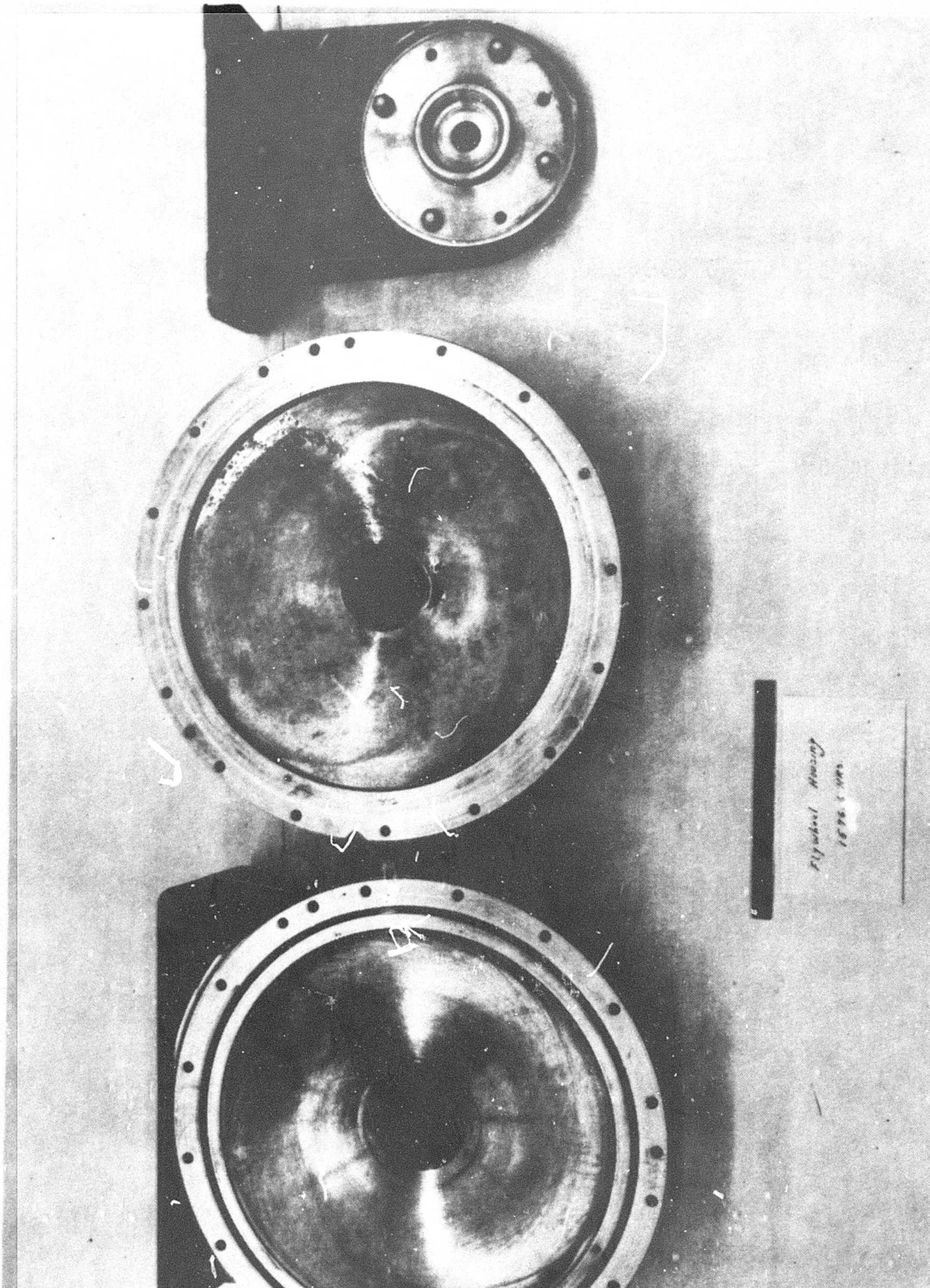


Figure 51. Closeup of Flywheel Housing

Figure 52. Flywheel After Continuous Duty Cycle Endurance Test

CARBON SEALS

The rotary carbon face seals and mating rings were found to be in functioning condition. The spring load of the carbon ring against the mating ring sealing face had not degraded and no appreciable amount of material appeared to have been worn from either part. The mating ring had solid residue, consisting of oxidized lubricant, varnished to its outer edge and on part of the sealing face. No part of either mating ring, however, displayed surface discoloration from metal oxidation. Figure 53 shows the sealing faces of both mating rings. The varnished lubricant can be seen coating the OD surfaces (lubricant side) and the scored grooves can be seen where the carbon seal rode. Inside of this, on the evacuated side, is a windrow of carbonized lubricant.

Although the localized temperature at the point of contact between the two parts was sufficient to decompose some of the lubricant, temperatures generally remained below that which would be necessary to damage elastomeric seals.

The parts (mating ring and carbon seal) were cleaned and inspected for surface finish on a profilometer. The surface condition was as follows:

	Surface Finish	Groove Depth
Inboard mating ring	20 to 50 RMS	.0005 inch
Inboard shaft seal	20 to 80 RMS	.0005 inch
Outboard mating ring	20 to 90 RMS	.0006 inch
Outboard shaft seal	15 to 90 RMS	.0008 inch

This roughened surface is believed to have been brought about by foreign particles, including metallic chips and other miscellaneous abrasives, which were introduced into the lubrication system at the time of initial assembly and/or generated during operation.

Apparently the contamination particles worked their way across the sealing faces and initiated the circular score marks. However, due to the ability of the carbon to conform to the convolutions in the mating ring face, leakage characteristics were not seriously affected. The carbon sealing ring is shown in figure 54.

SPLINED QUILL SHAFT

The quill shaft, which interconnects the flywheel and the gearbox pinion, was designed as a symmetrical part. The end which inserts into the flywheel engaged the full length of the mating spline. However, the opposite end, which inserts into the gearbox pinion, engaged only about 80 percent of its spline length due to the configuration of the mating part.

No wear was apparent on the fully engaged spline. However, wear on the other end had removed approximately 50 percent of the teeth cross sections. The female mating part (i.e., the gearbox pinion) displayed about an equal amount of wear. A brown paste wear particle residue typical of

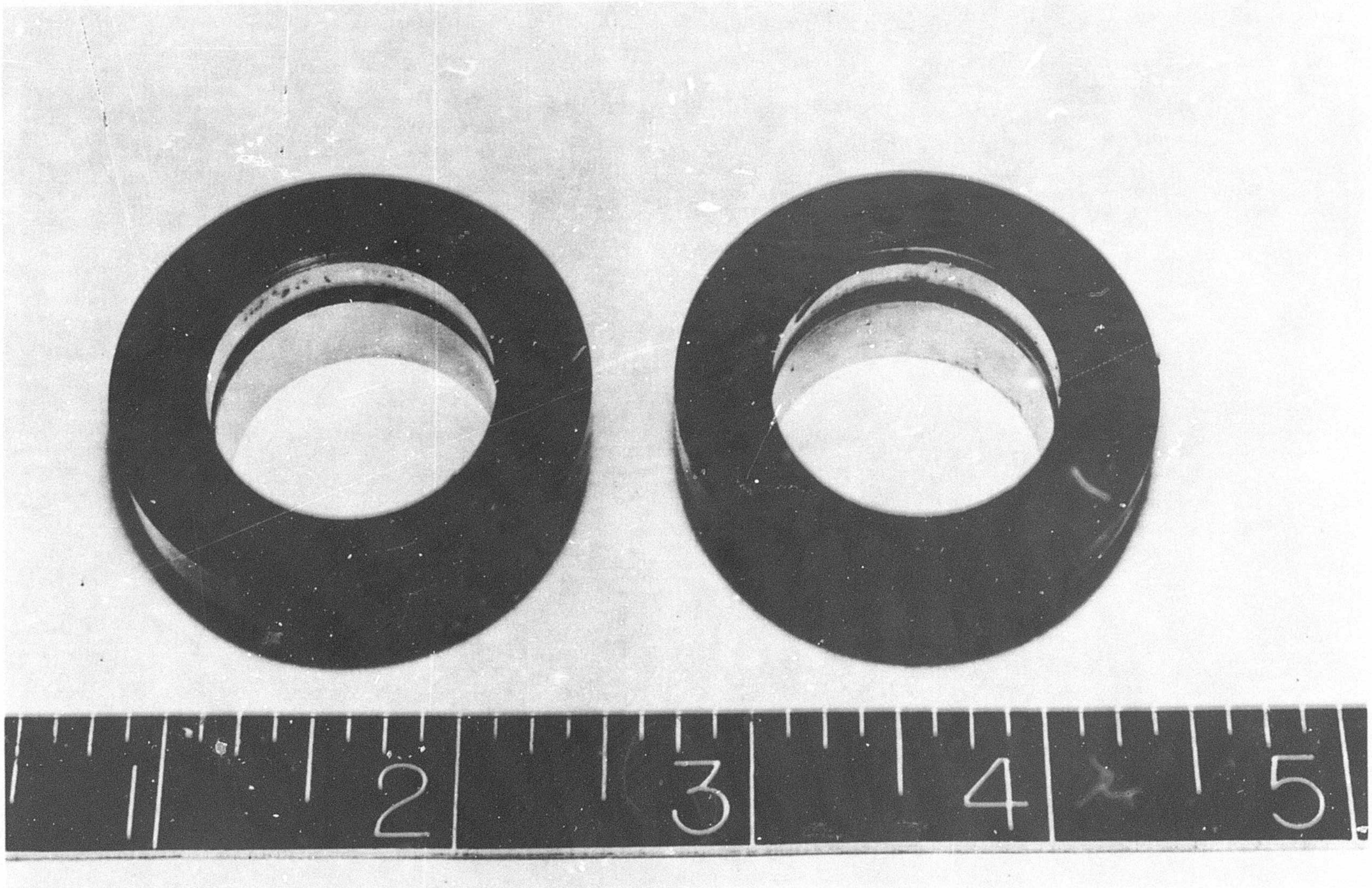


Figure 53. Flywheel Seal Mating Rings (Relapped Prior to 1339.9 Hr Run)

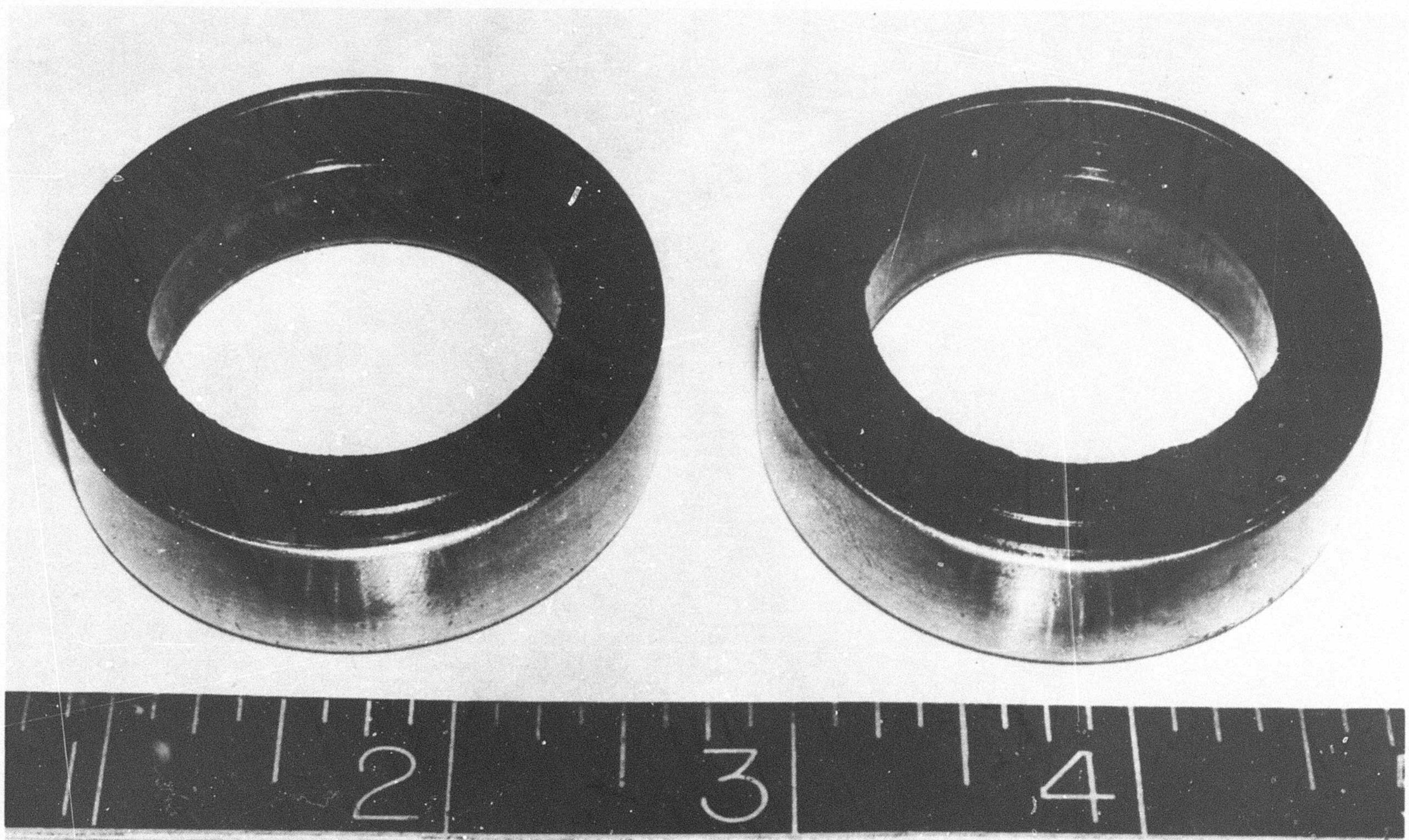


Figure 54. Flywheel Shaft Seal

"fretting corrosion" had built up around the areas of contact. It is estimated that over 70 percent of the spline life had been used up by the test. Both the worn and unworn ends of the quill shaft can be seen in figure 55.

BEARINGS

The flywheel bearings appeared to be in excellent condition. The balls retained their polished brightness. The ball pockets in the retainer showed no signs of elongation or distress. There was no sign of overheating and to all appearances, the lubrication system had performed well. There were some scratches and scuff marks on the balls and the bearings were noticeably rougher than when originally installed. A photograph of both inboard and outboard bearings is shown as figure 56.

GEARBOX

Before disassembly, examination of the gearbox showed that all parts were functioning freely. The gearbox could be hand rotated with noticeably less drag than before the start of the test, and there was no audible sound. The unidirectional brake (overrunning clutch) was also functioning properly.

Disassembly revealed all internal parts to be clean, and there were no visual signs of extraordinary wear except at the service pump group driving shaft. The service pump group is driven at motor-pump speed by a shaft which uses a rectangular key in a slot as a driven spline. The driving sides of the slot were worn to a 30-degree slope. The driven key was worn in a corresponding manner until about one half of the thickness was worn through at the outer edge. However, shearout would probably not have occurred for several thousand hours. The worn key and slot is shown in figure 57.

The completely disassembled gearbox and service pump group is shown in figure 58. It can be seen that the general appearance of all parts ranges from fair to excellent after 1,576.6 hours of operation. A closer view of the gears and their associated bearings, as they would appear looking from the motor-pump side with half of the housing removed, is shown in figure 59. A corresponding view, looking from the other direction, is shown in figure 60.

The motor-pump side view of figure 59 shows that, in spite of the general excellent appearance, certain areas are starting to show signs of distress. These are the motor-pump input spline area and the high speed pinion support bearings. Both of these show signs of overheating.

A closeup of the motor-pump input spline area is shown in figure 61. It can be seen that the shaft seal has started to score the shaft and that the spline area has been overheated. The overheating probably occurred from the combined effects of seal heat generation, spline heat generation, and lack of sufficient lubricant flow in this area to carry the heat away.

Figure 62 is a magnified view of the high speed pinion and one of its support bearings. It can be seen that most of the silverplating is missing from the ball retainer and that both the balls and the retainer are darkened.

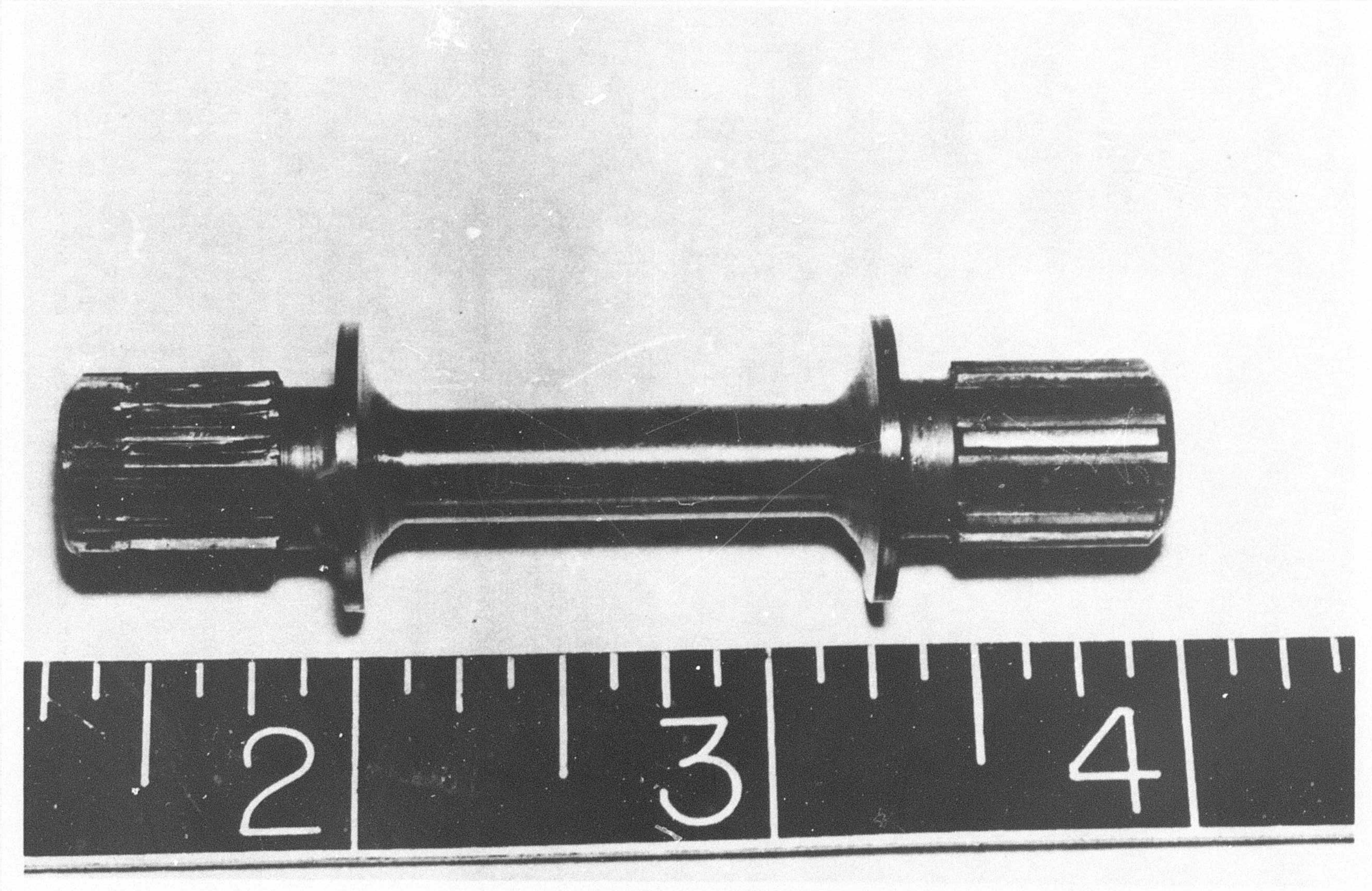


Figure 55. Flywheel - Gearbox Quill Shaft

Figure 56. Inboard and Outboard Flywheel Bearings

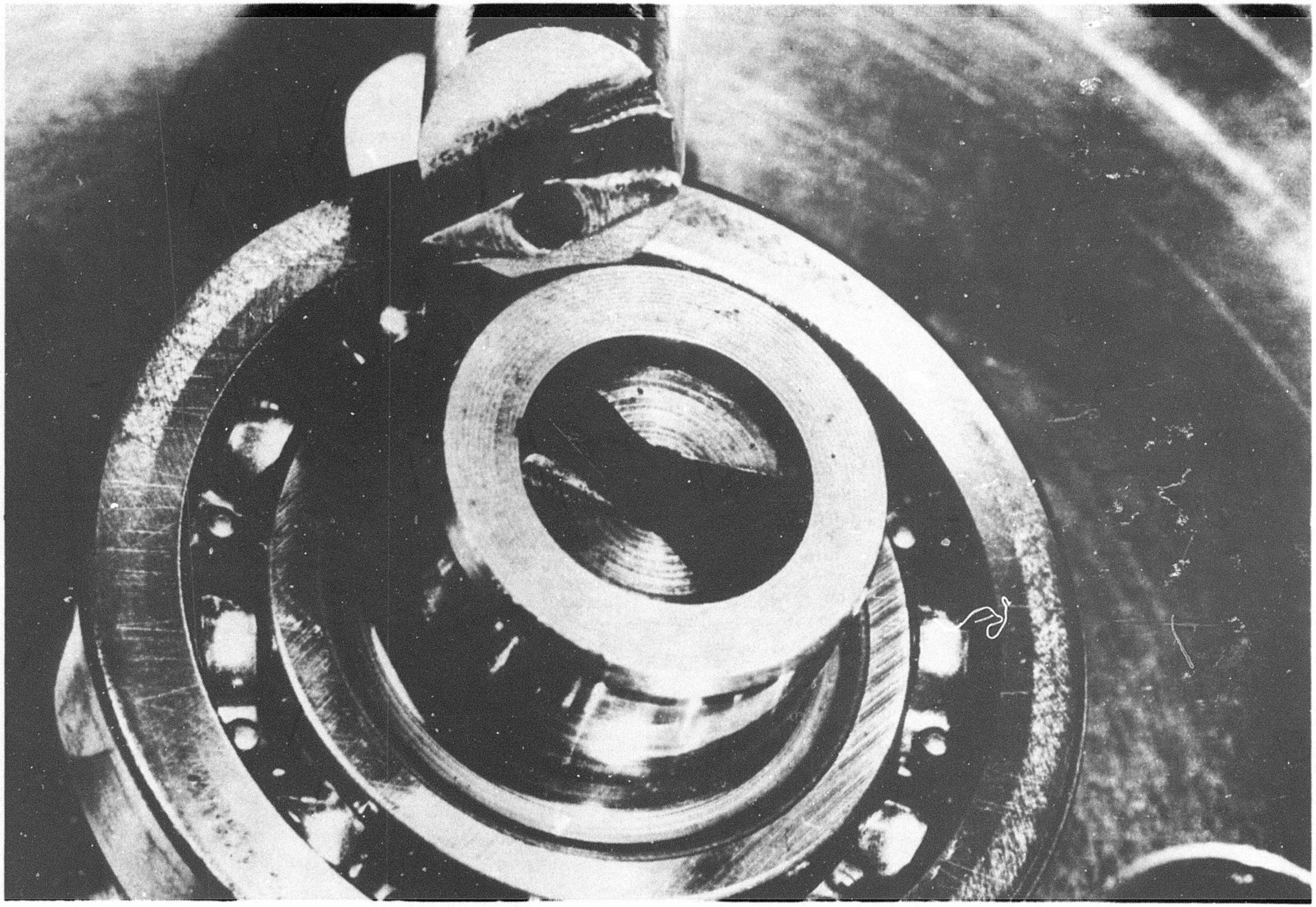


Figure 57. Service Pump Group Drive Wear

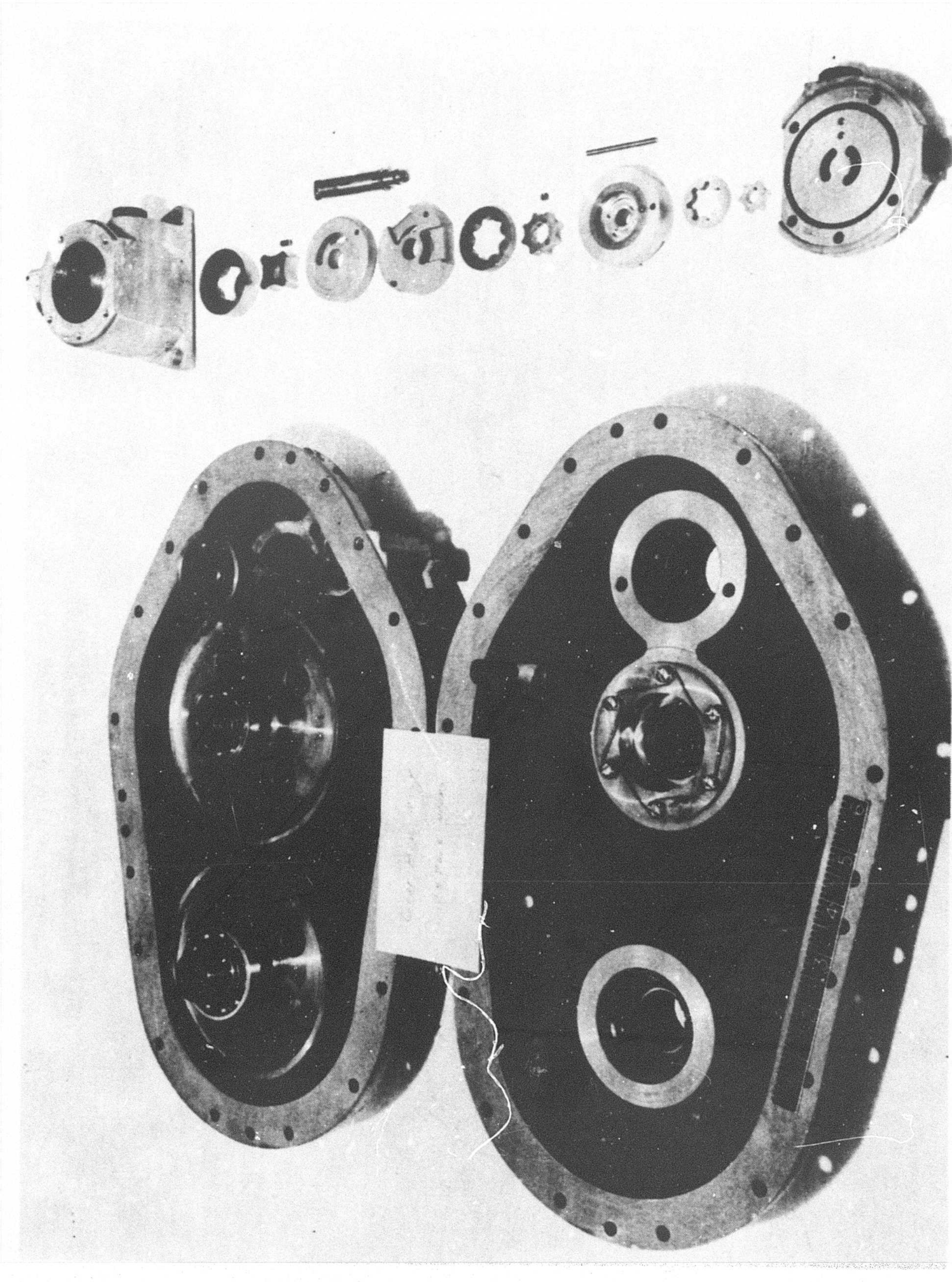


Figure 58. Gearbox and Service Pump Group Disassembled

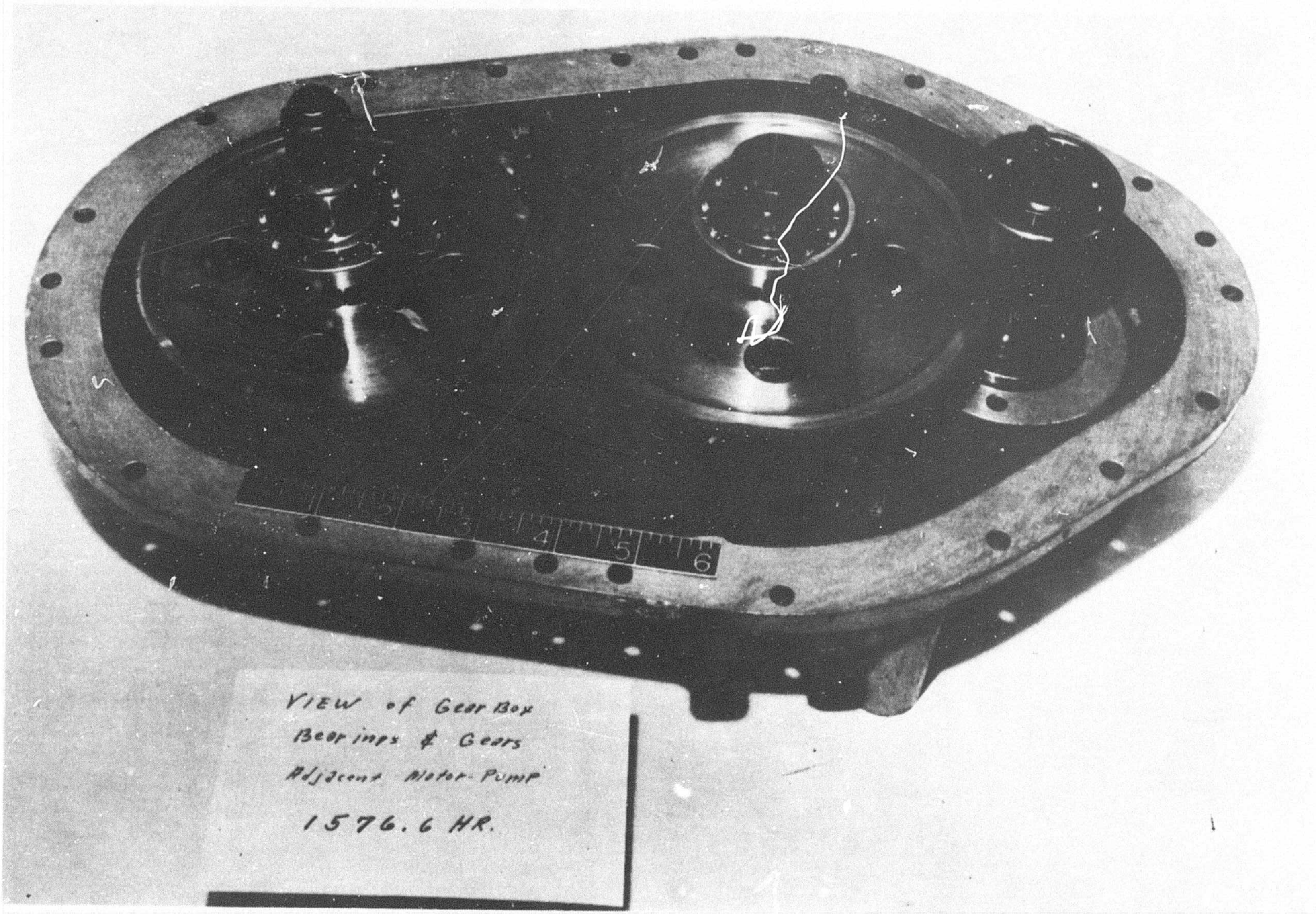


Figure 59. View of Gearbox Half From Motor Pump Side

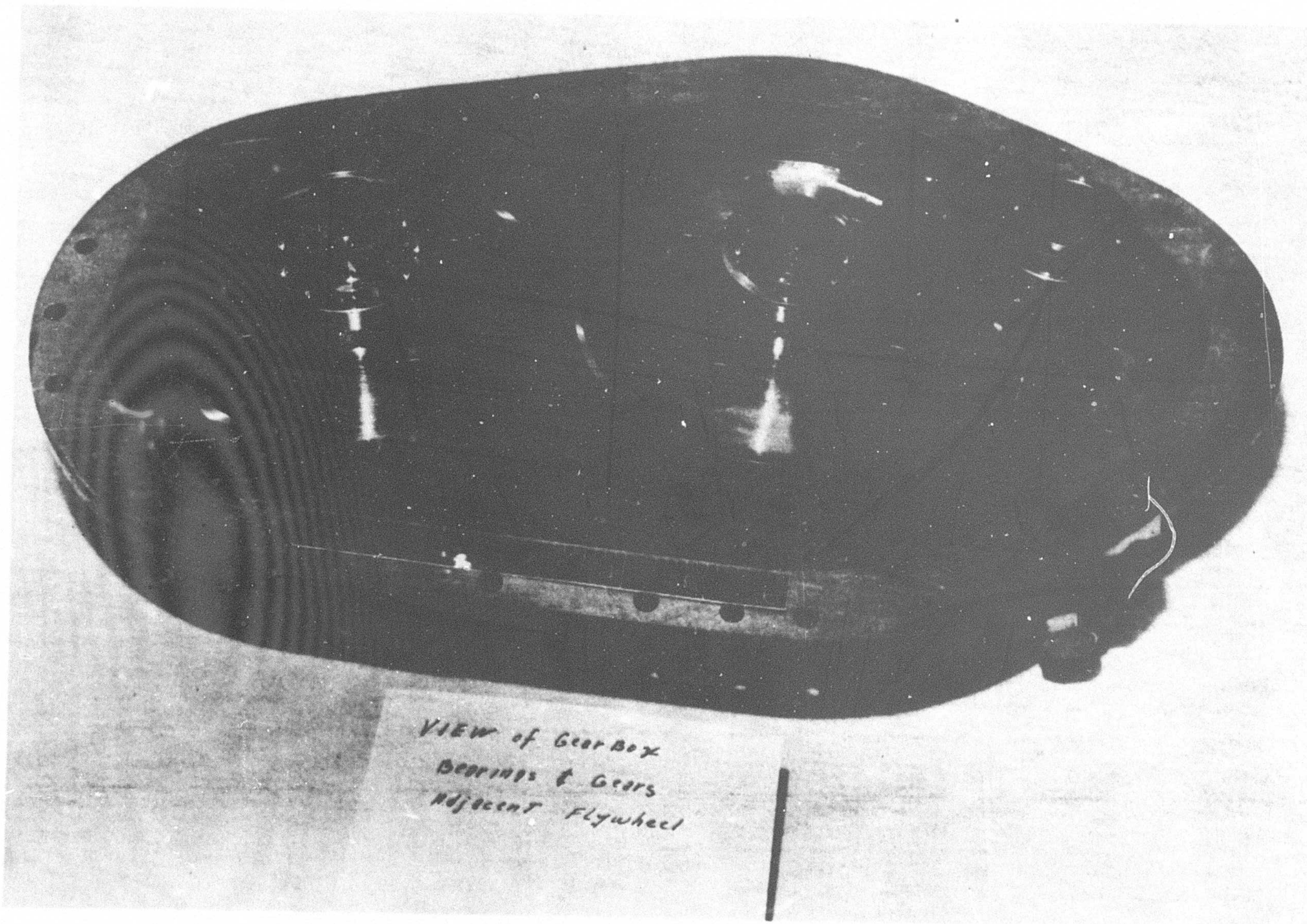


Figure 60. View of Gearbox Half From Flywheel Side

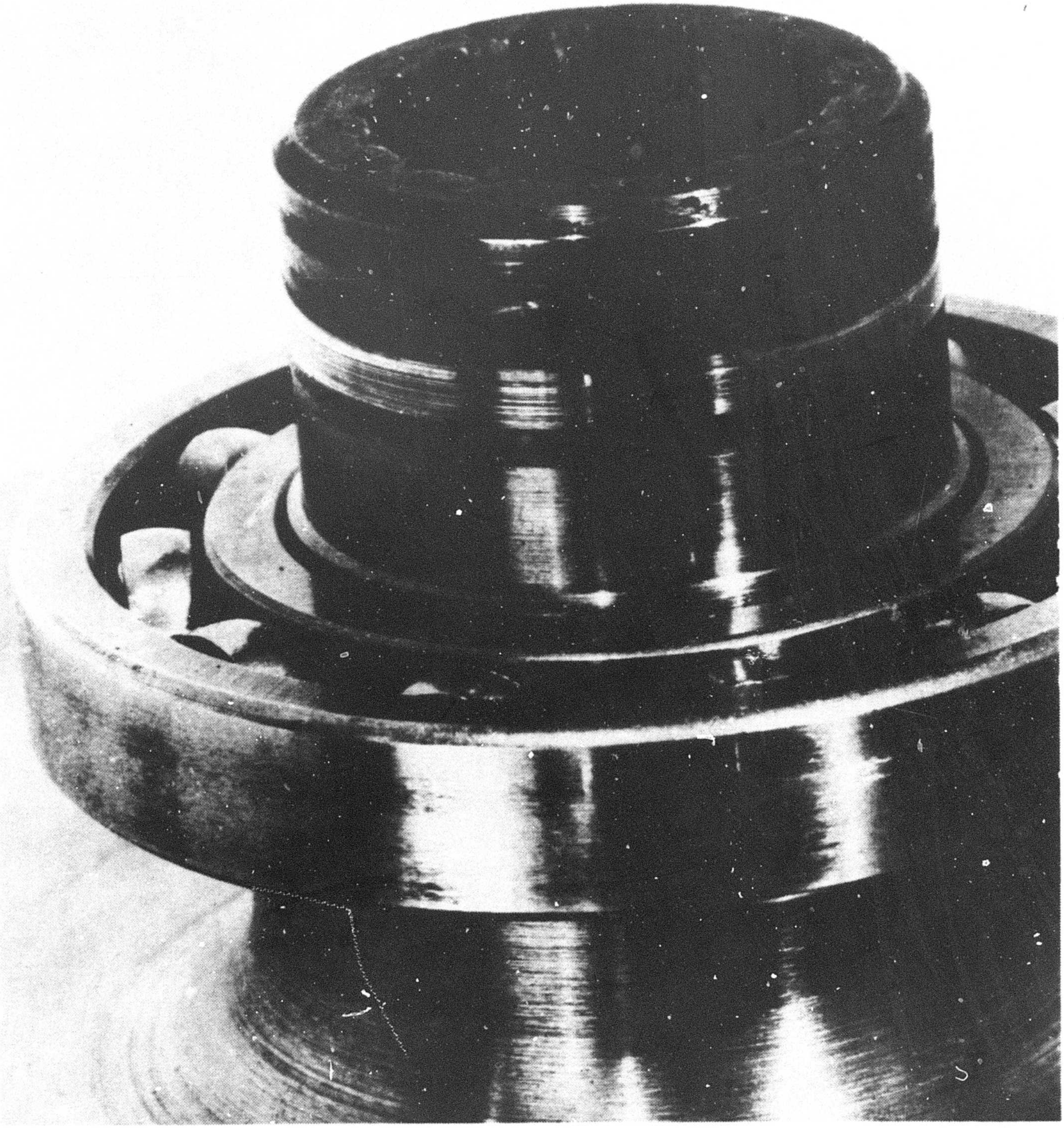


Figure 61. Gearbox Spline - Input From Motor Pump

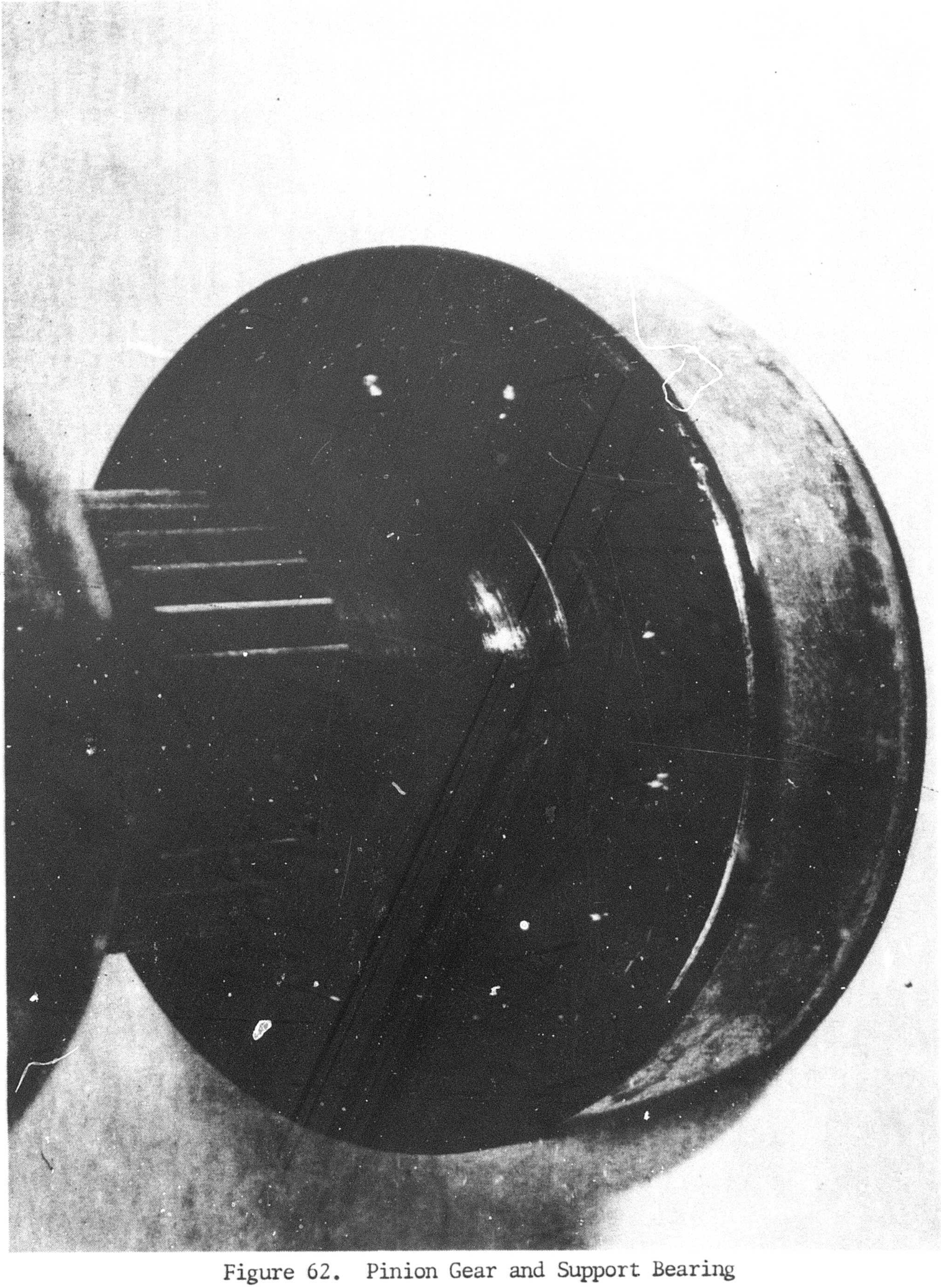


Figure 62. Pinion Gear and Support Bearing

Although it cannot be seen in this photograph, the balls were straw-colored and the ball retainer was blued. This was true of both bearings. In contrast to this, the pinion gear was in excellent condition.

Figure 63 compares the pinion support bearing to an identical flywheel bearing which operated adjacent to it in the flywheel housing at the same speed. The contrast, resulting from heating of the pinion support bearing, is apparent. In most respects the operating conditions for the flywheel bearing were more severe; however, the flywheel bearing was jet lubricated whereas the pinion support bearings were mist lubricated. From this it is apparent that jet lubrication for this type bearing, operating under these conditions, is much to be preferred.

It has already been pointed out that the high speed pinion teeth were in excellent condition. However, both of the larger gears showed signs of distress. Figure 64 is a closeup of the hydraulic motor-pump-driven gear's teeth. The appearance of slight spalling can be seen on the flank of one of the teeth slightly below the center of the picture. Figure 65 is a magnified view of this particular tooth and shows that the spalling is quite widespread. Although a tooth profile reading was not taken, it is believed that the spall depth did not exceed .0004 inch.

All three W. H. Nichols Gerotor pumps in the service pump group were in excellent condition. The disassembled vacuum pump, lube pump, and scavenge pump are shown in that order in figures 66, 67, and 68, respectively.

Visual teardown inspection of the New York Airbrake motor-pump revealed that the unit was in excellent condition considering the 1,523.6 hours of operation under a cyclic pump-motor duty cycle. The disassembled unit is shown in figure 69. The following wear conditions were noted:

1. Upon disassembly the needle bearings used in the wobble plate trunnion were not retained and fell out. The trunnion exhibited some marking from the needle bearings. The needle bearing retention mechanism was missing.

2. Normal valve plate wear, with some cavitation erosion adjacent to the kidney porting V-notch, was evident (see figure 70). Valve plate wear on the mating piston barrel consisted of circumferential grooving of the running surfaces (see figure 71).

3. Slight localized erosion of the slipper (wobble) plate at two points 180 degrees from each other were noted (see figure 72). The shape of this erosion could be matched with the outline of the piston slipper shoes. An enlarged view of the most apparent erosion is shown in figure 73.

4. The mode change mechanism exhibited some abrasive wear at the ball joint which mates with the wobble plate. The nature of the wear is shown in figure 74.

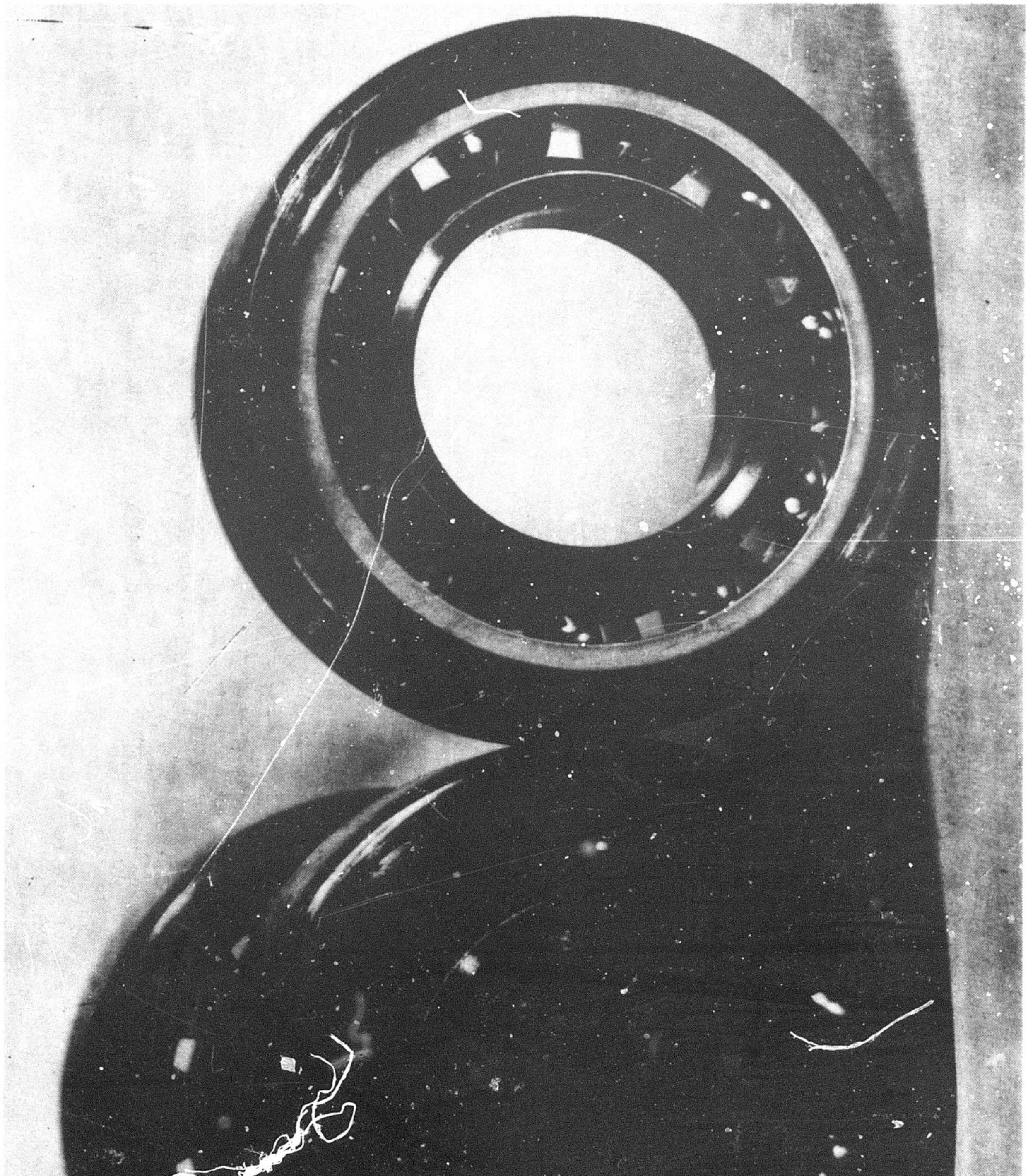


Figure 63. Comparison of High Speed Support Bearings

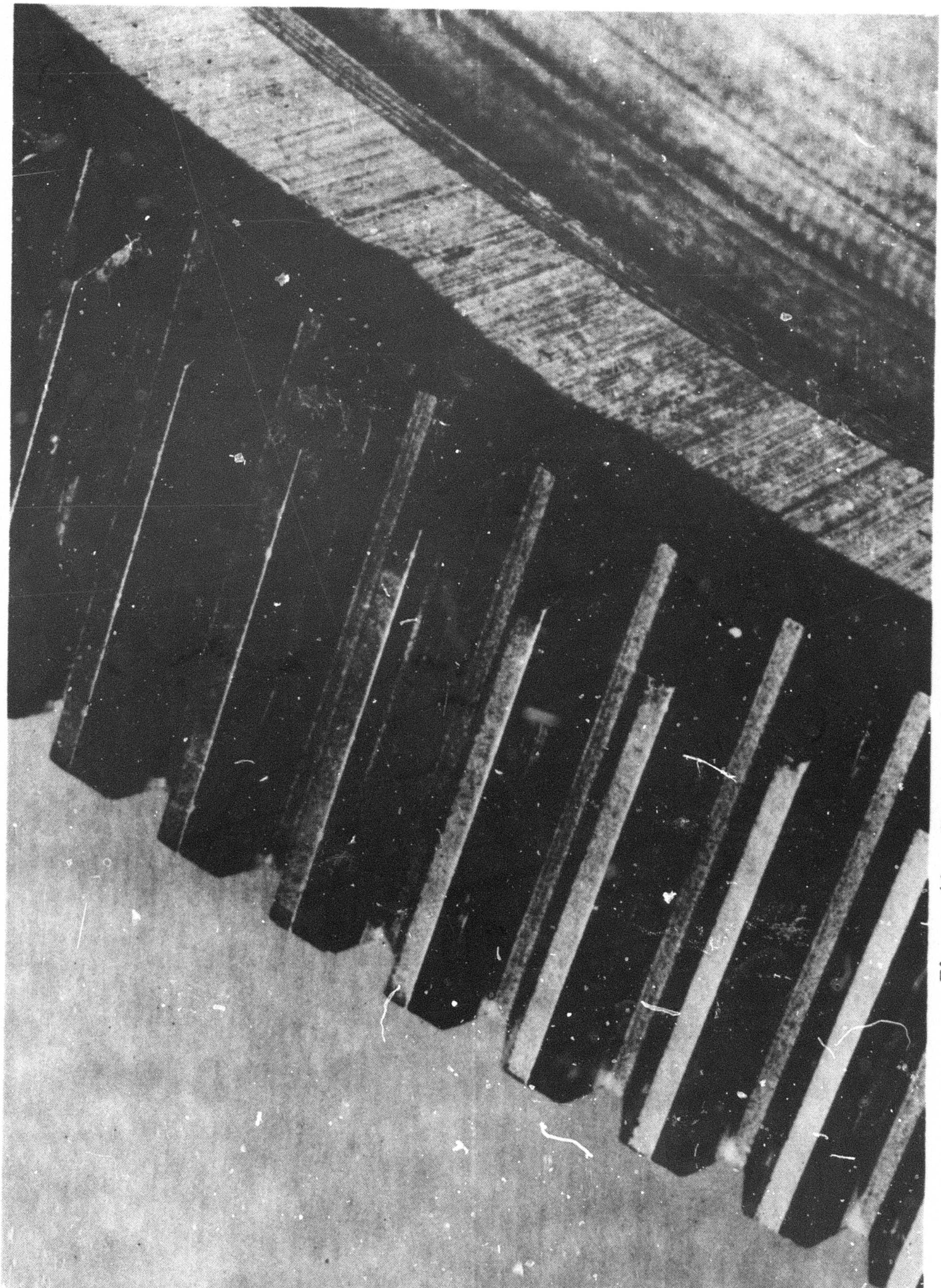


Figure 64. Closeup of Motor-Pump Driven Gear

Figure 65. Motor-Pump Driven Gear Tooth Magnified

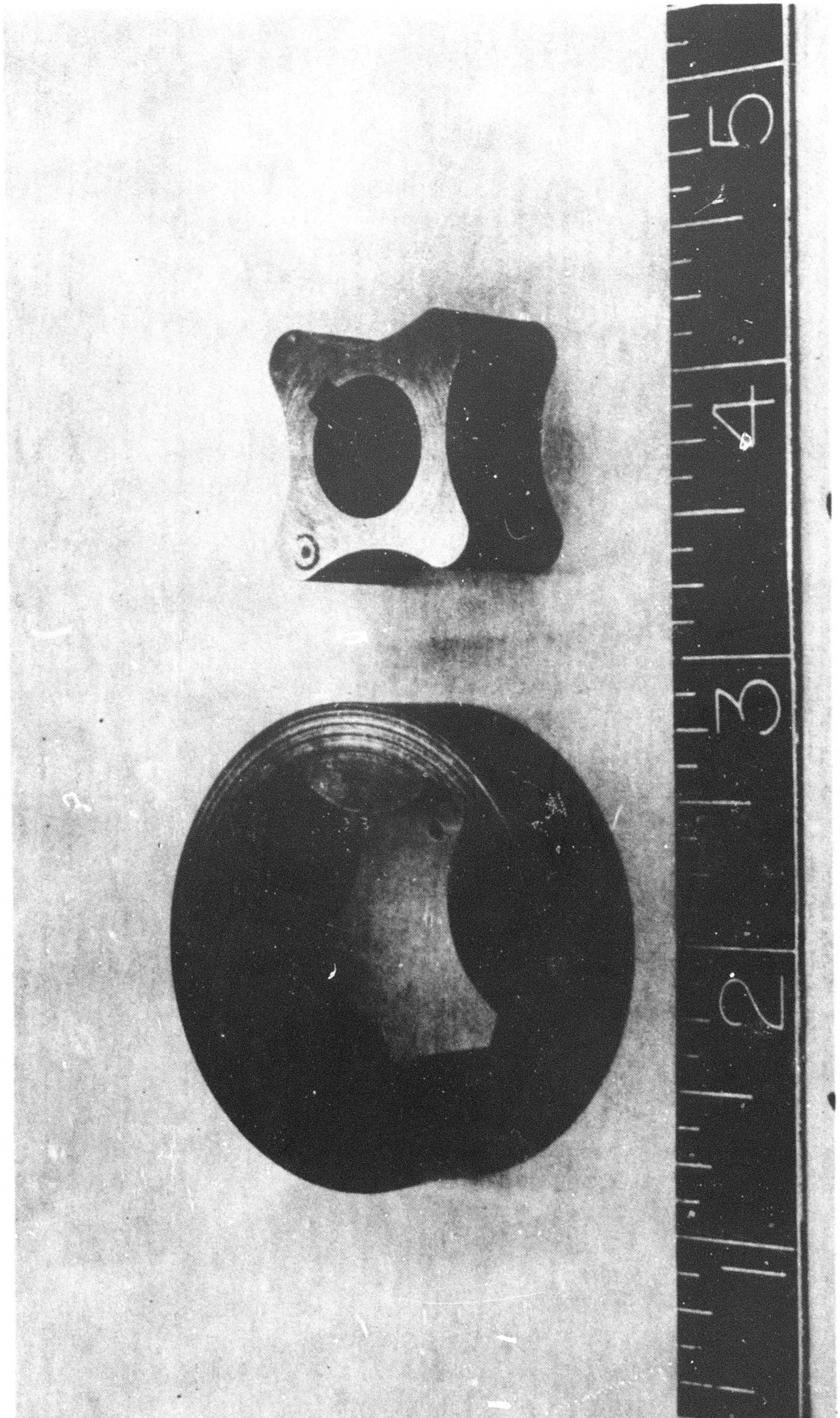


Figure 66. Gerotor Vacuum Pump

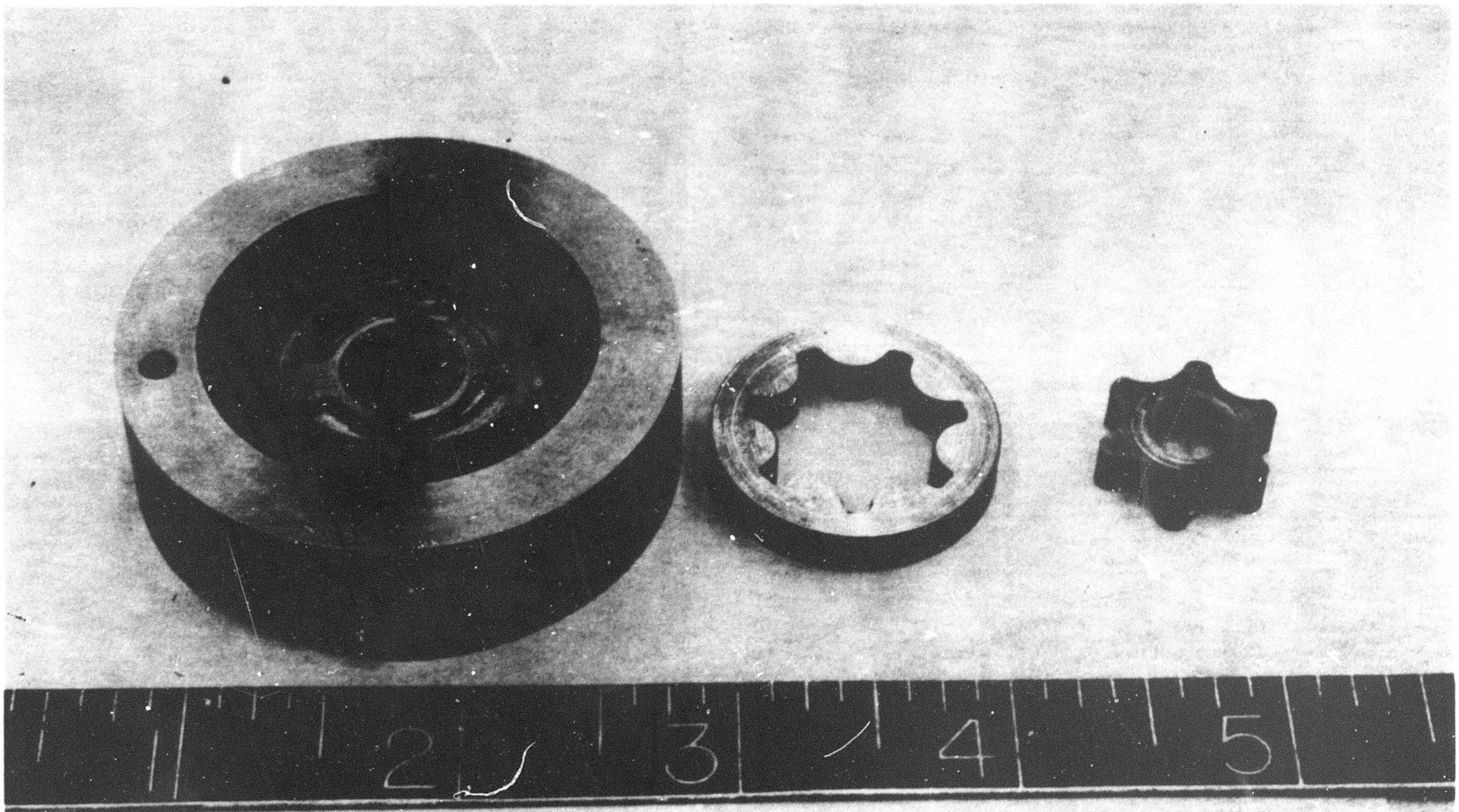


Figure 67. Gerotor Lube Pump

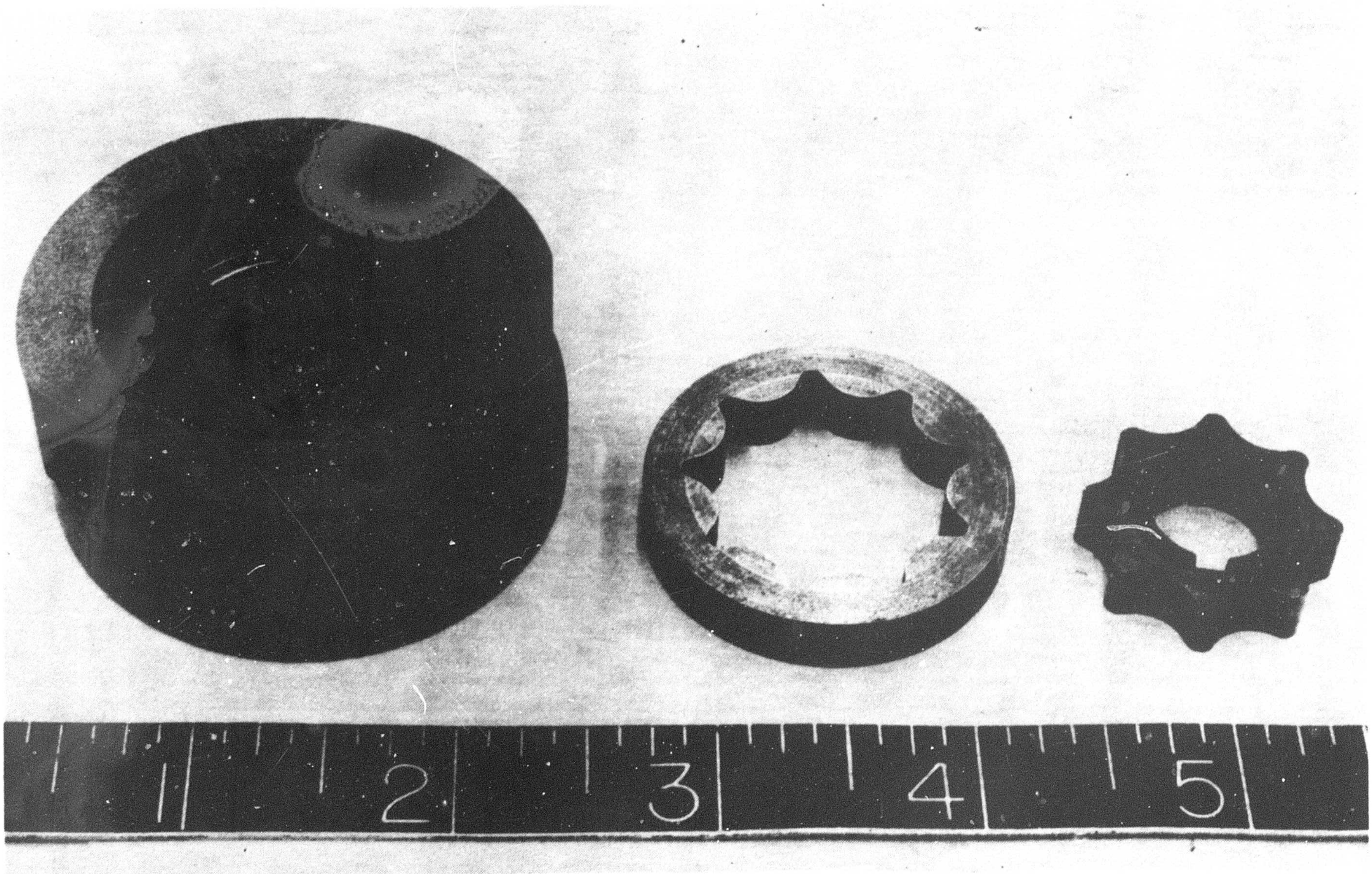


Figure 68. Gerotor Scavenge Pump

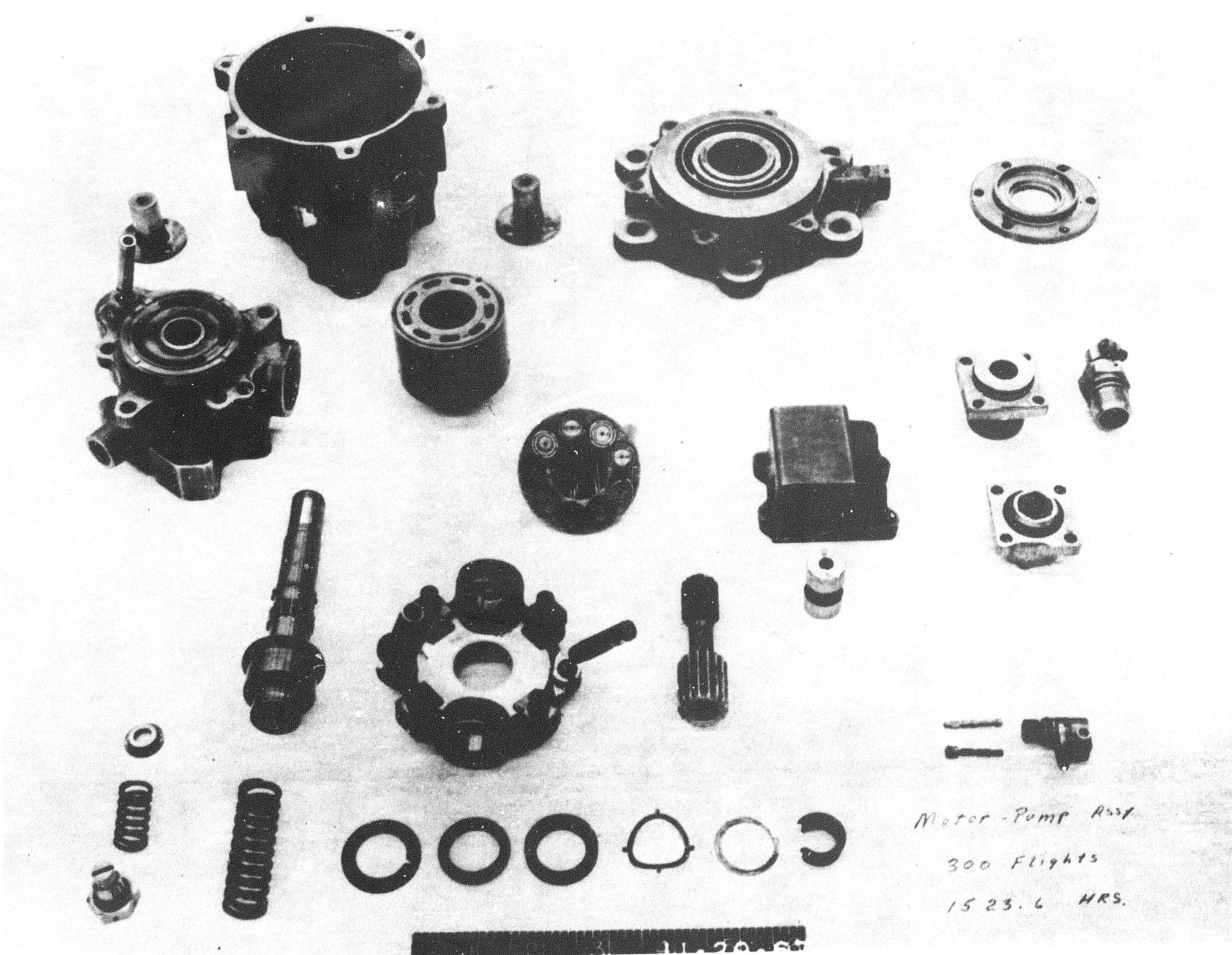


Figure 69. Disassembled Motor-Pump

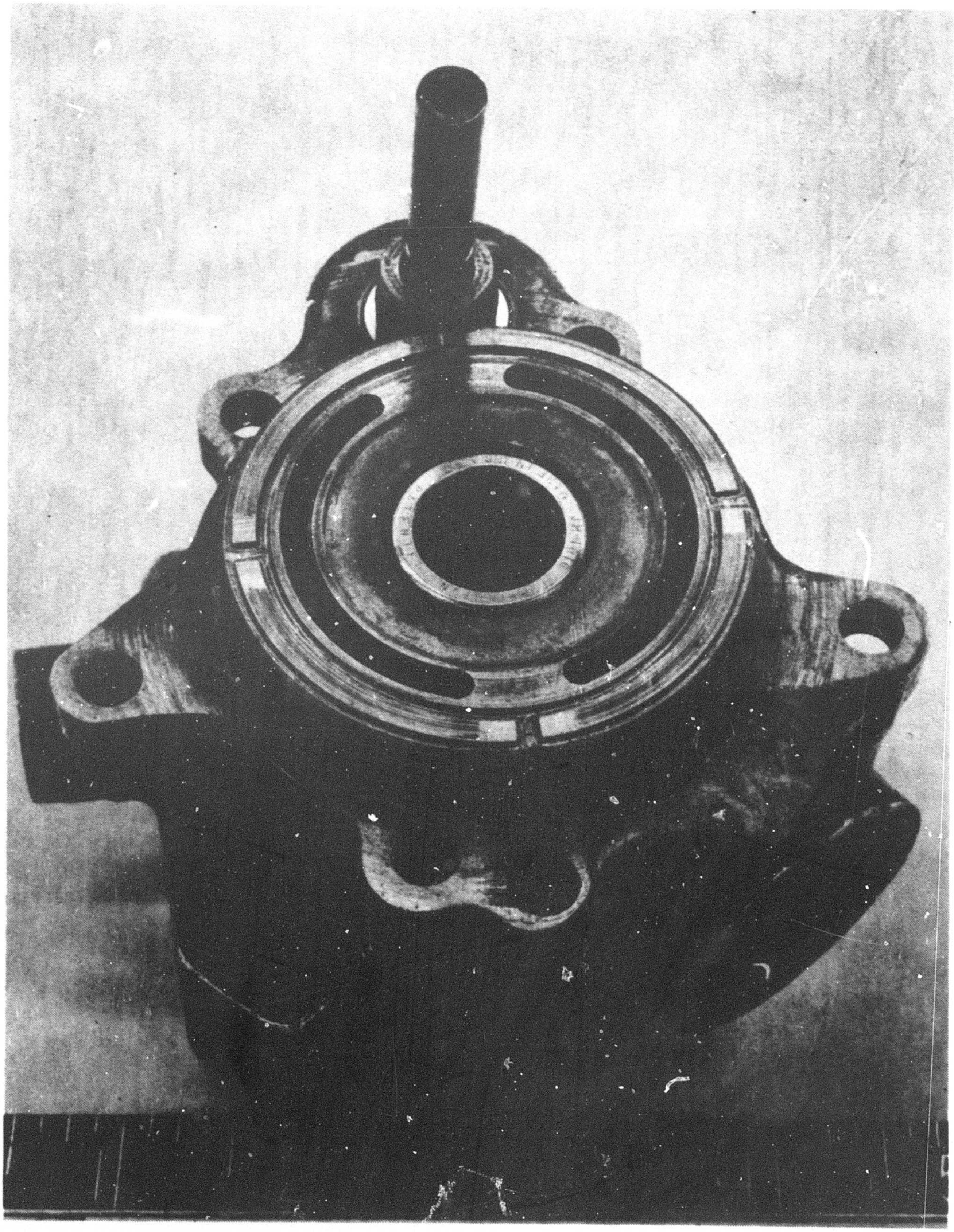


Figure 70. Motor-Pump Kidney Port Wear

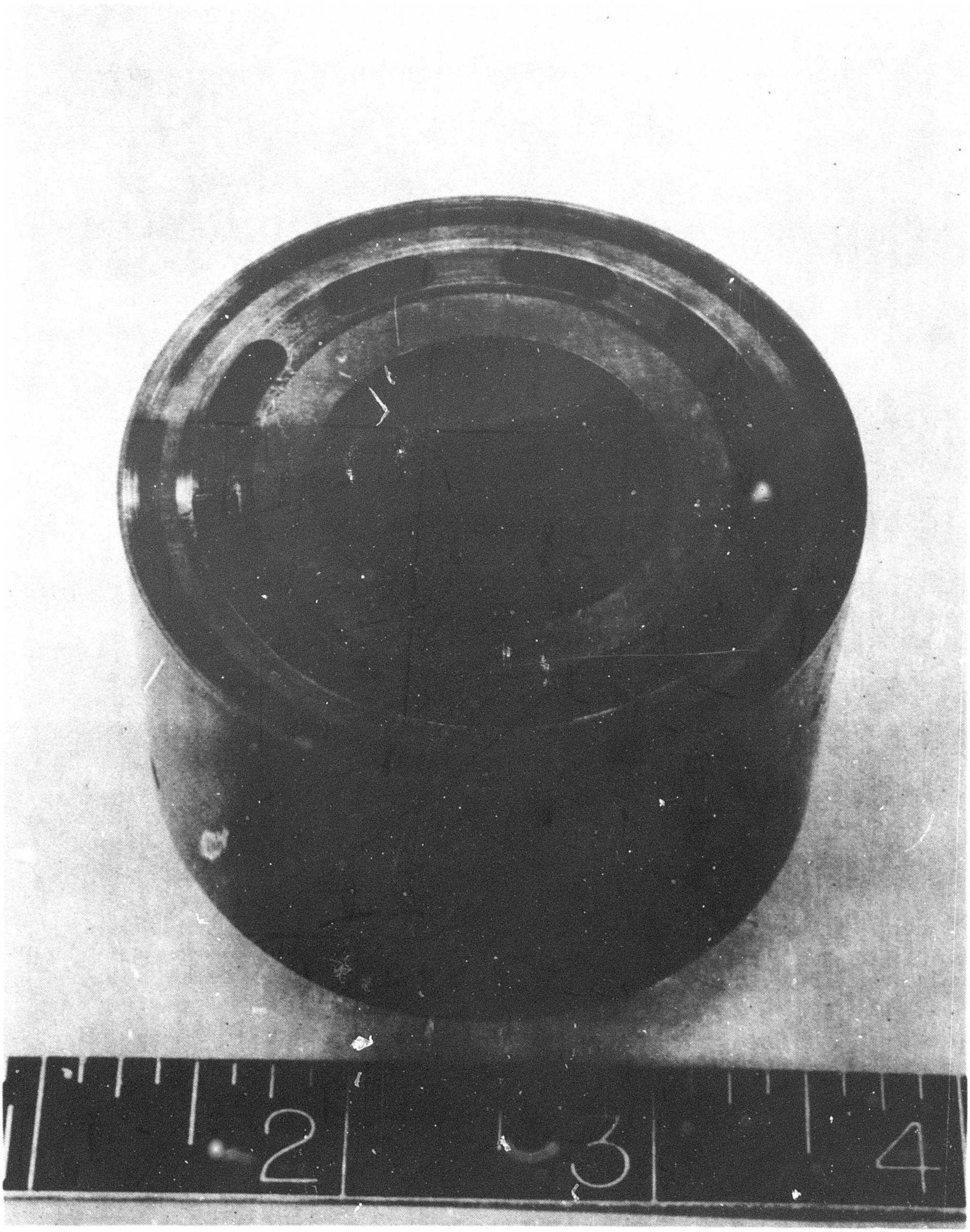


Figure 71. Motor-Pump Piston Barrel Valving Surface Wear

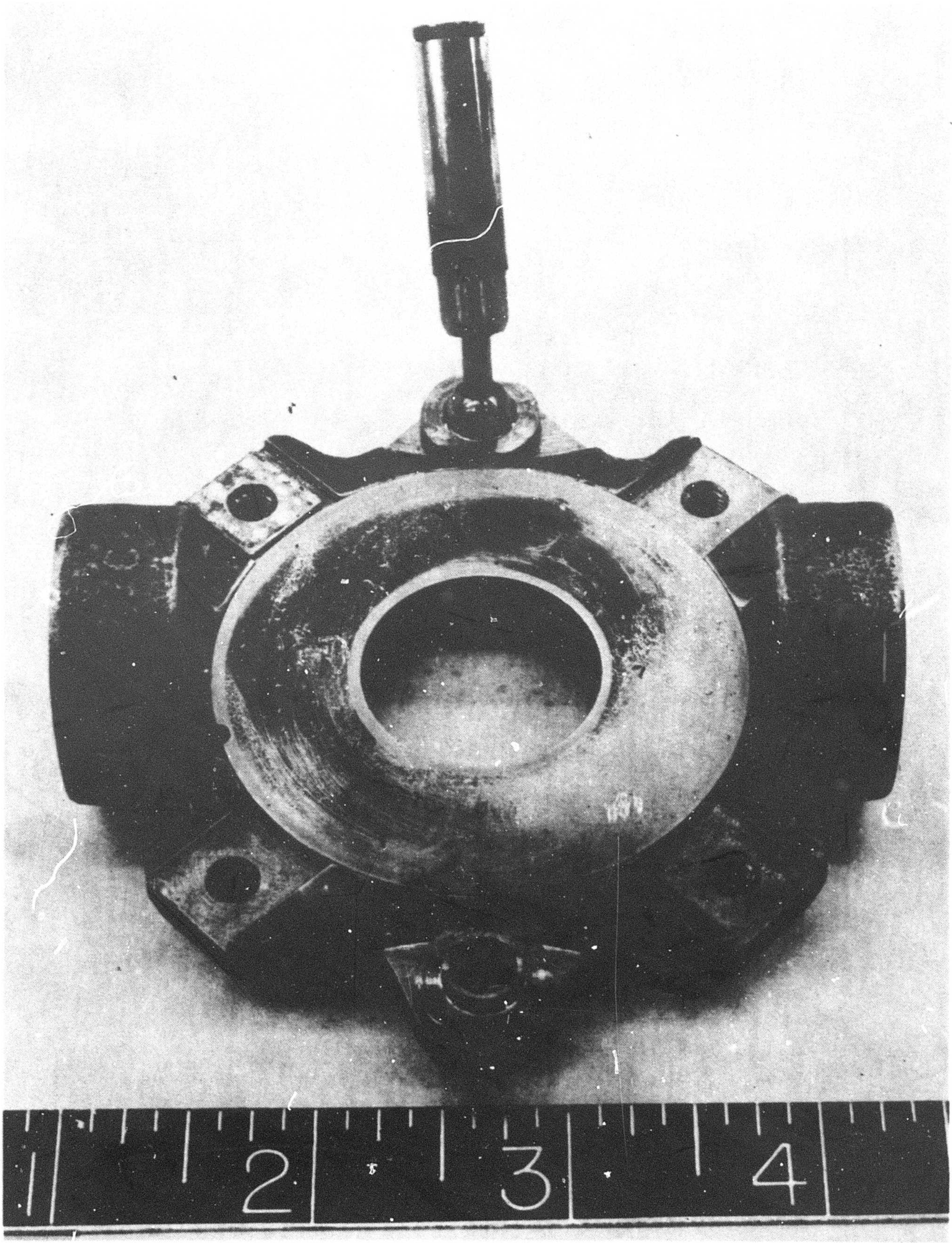


Figure 72. Motor-Pump Wobble Plate

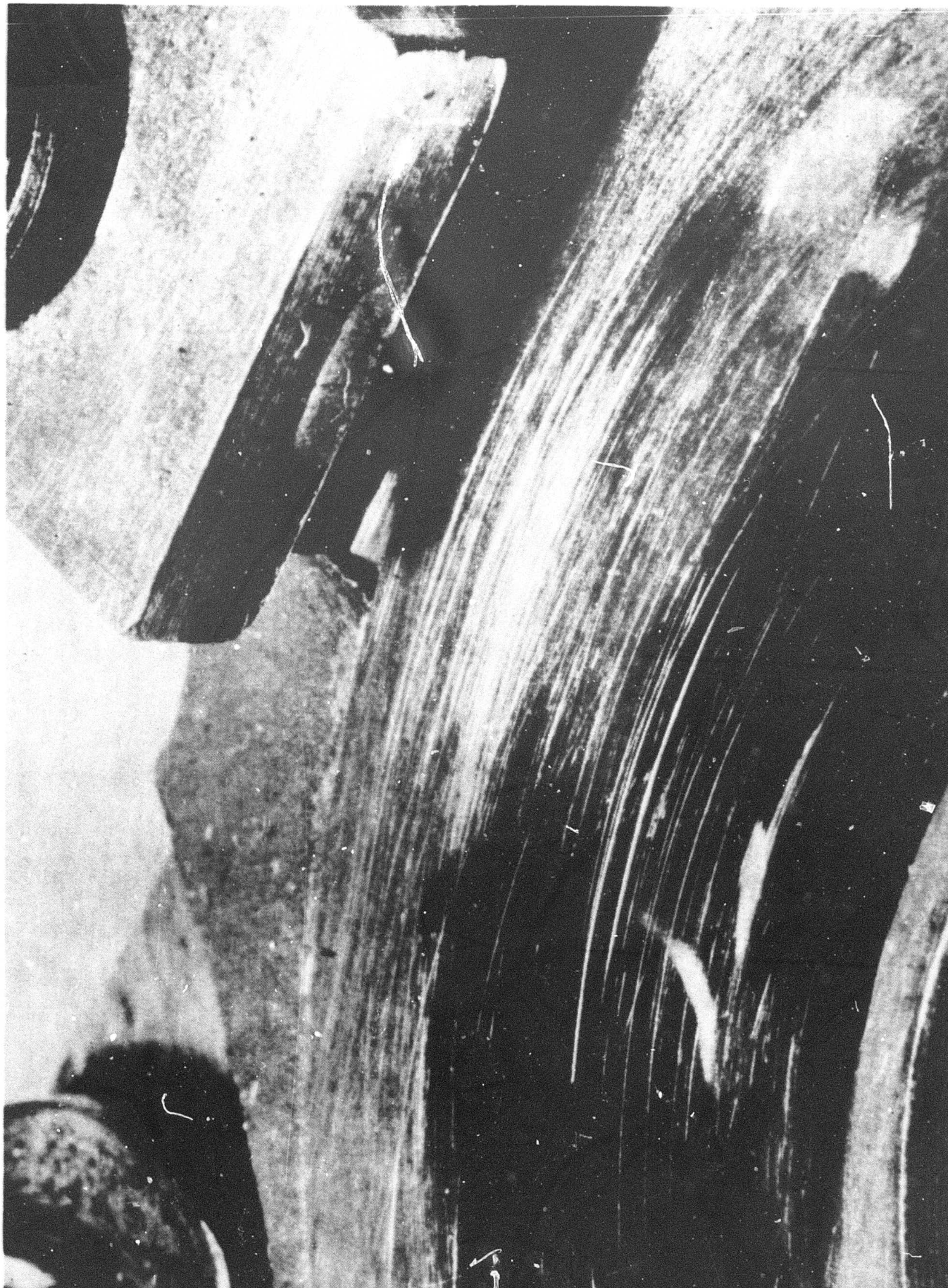


Figure 73. Closeup of Wobble Plate Erosion

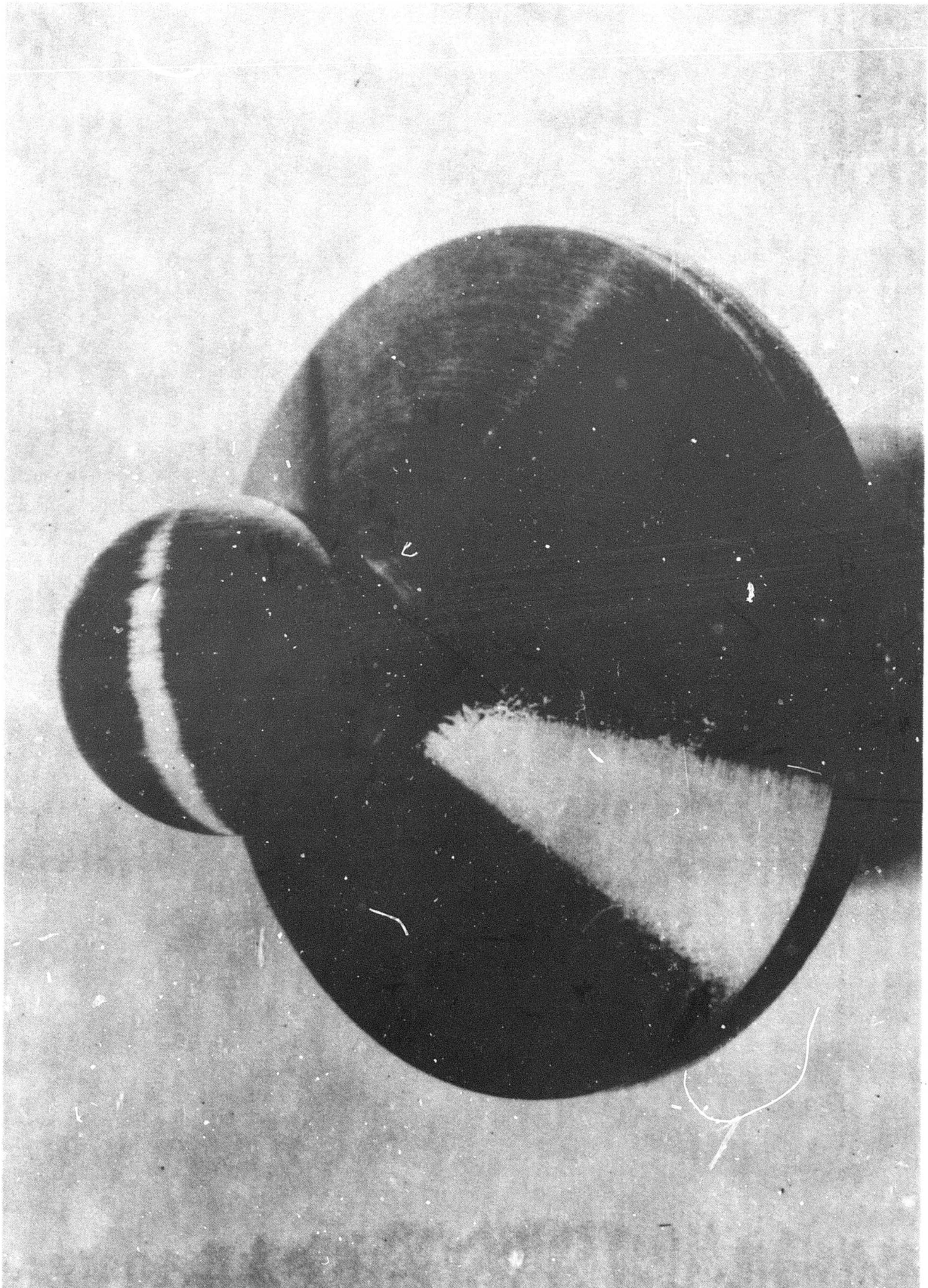


Figure 74. Motor-Pump Control Actuator Ball Joint

5. Erosion was noted at the outer diameter of the piston slipper shoes. This is shown in figure 75 and the enlarged view of figure 76.

6. The control valve shift spool was badly eroded on the large diameter of the spool. The severity of this is shown in figure 77.

7. The pump control actuator piston ball joints were found to have greater than normal freeplay.

The excessive erosion found on the shift spool and the outer diameter of the piston slipper shoes account for the contaminant found in the filters on initial runs of this motor-pump assembly. Increased case pressure at this time resulted in elimination of pump-generated contaminant. There was no pump-generated contaminant found after the increase of case pressure, indicating that the erosion process had been stopped.

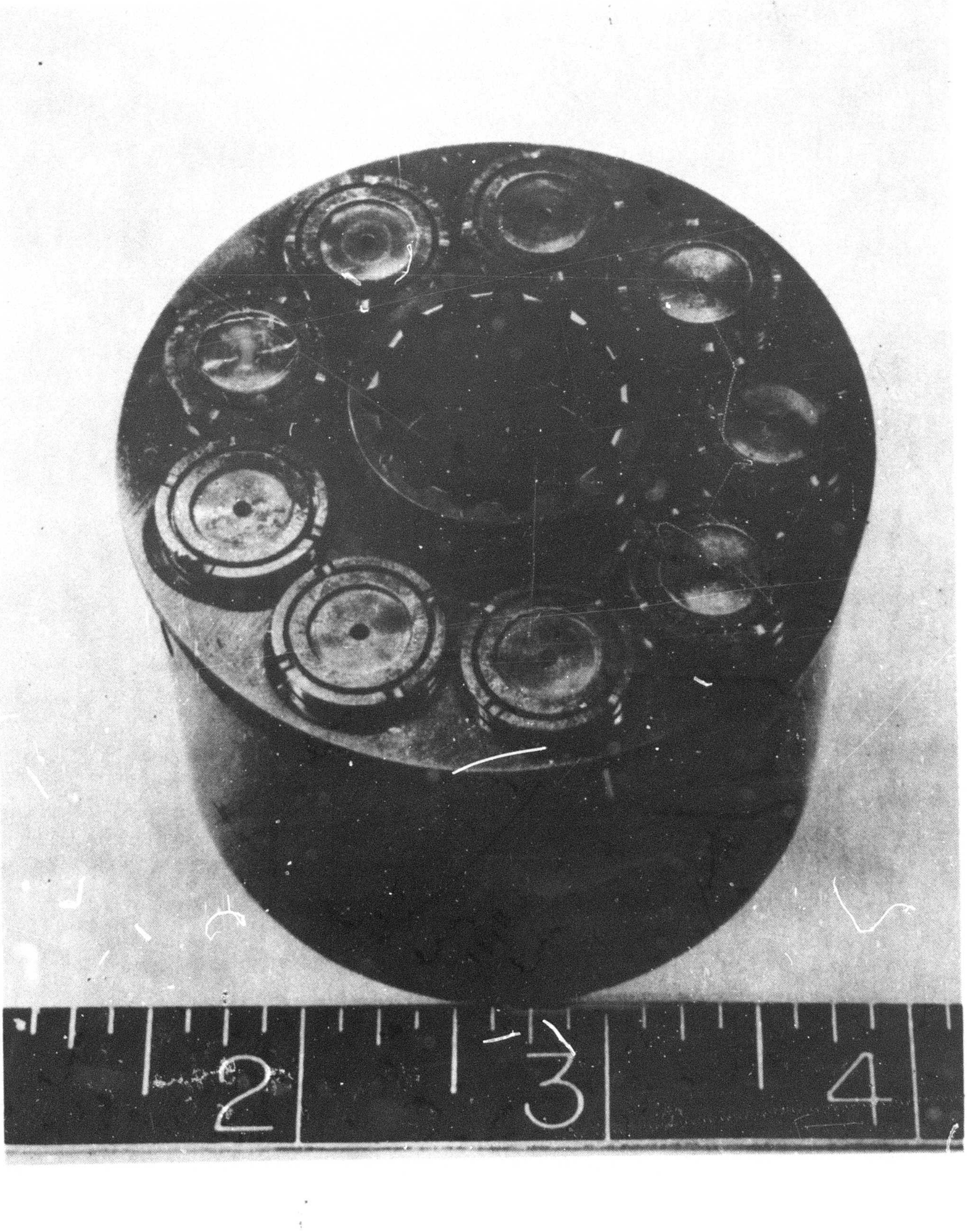


Figure 75. Motor-Pump Piston Slippers

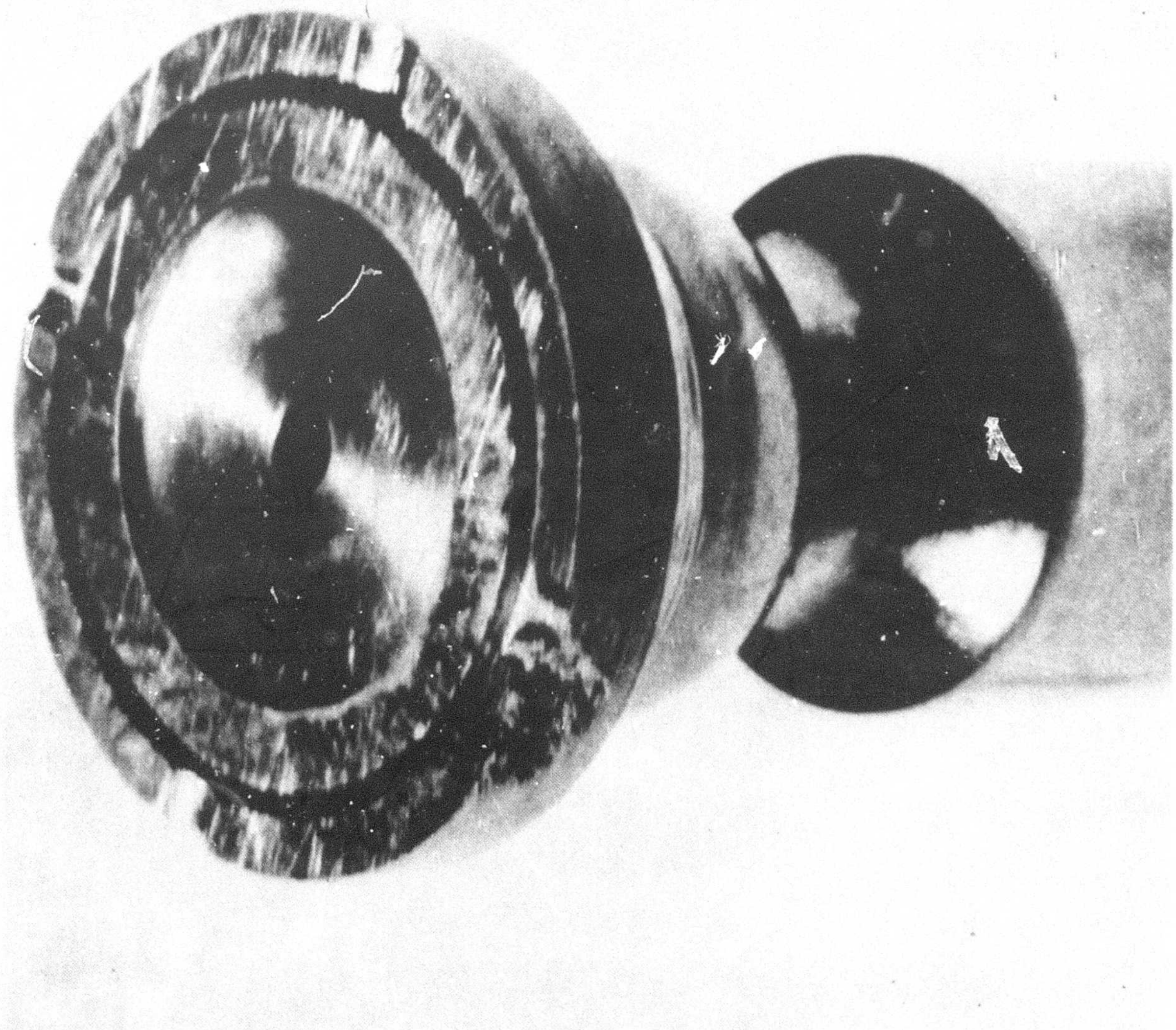


Figure 76. Enlarged View of Slipper Erosion

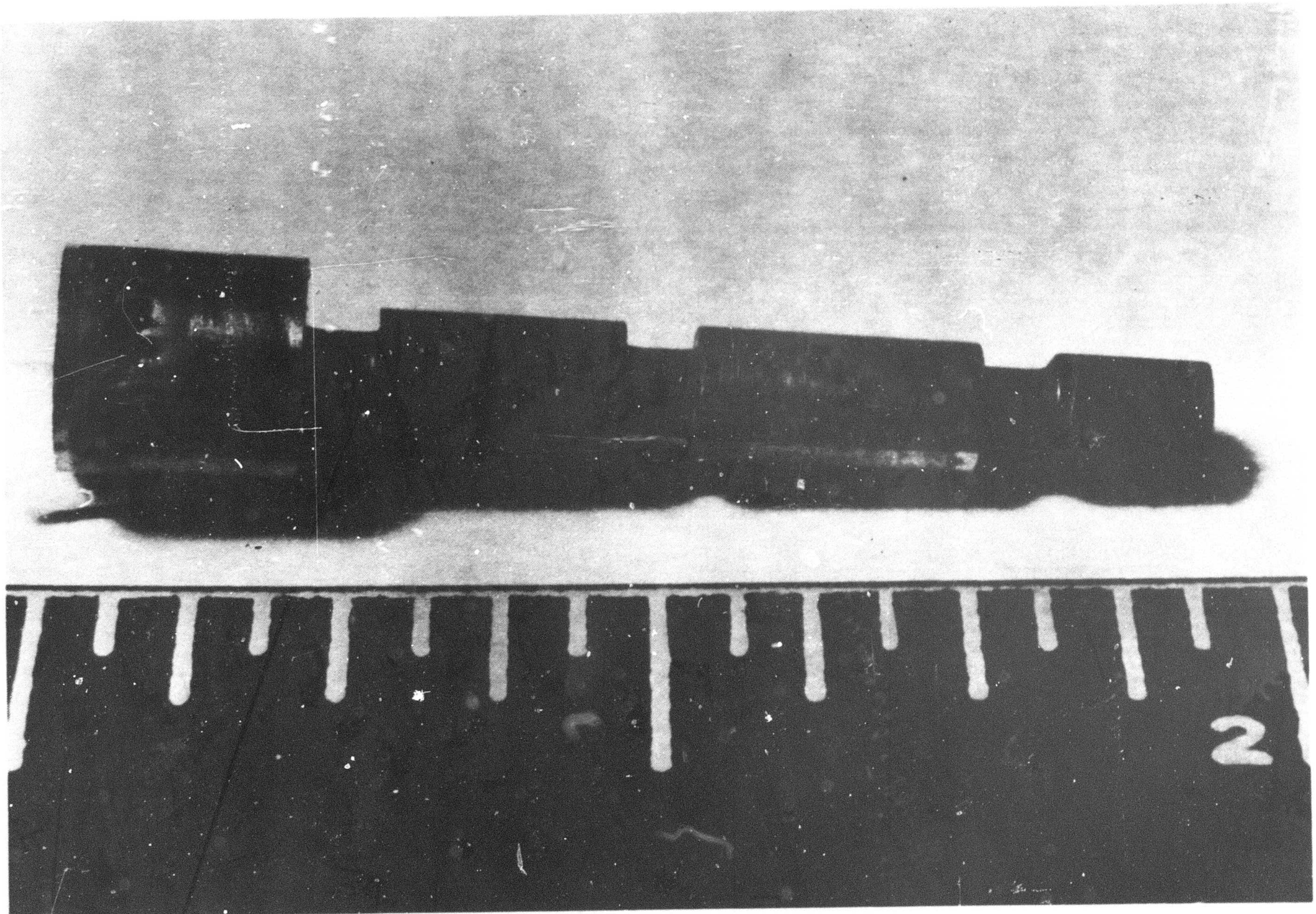


Figure 77 Motor-Pump Shift Spool Erosion

Section VI

PROGRAM SUMMARY AND CONCLUSIONS

Of the four basic demonstrations which constituted this program, two were very sucessful and the other two indicated that additional work should be done. The demonstrations conducted on the hydraulic version of the intermittent duty cycle simulator and the continuous duty cycle simulator were very sucessful. The former completed 10,000 landing gear cycles without an incident directly chargeable to the substation. The latter completed 1,500 hours on a typical flight control duty cycle and probably could have completed an additional 500 hours.

The tilt table test demonstration and the demonstration conducted on the mechanical version of the intermittent duty cycle simulator were less successful. The former was targeted at demonstrating the equivalent of 3,000 hours of life for the flywheel bearings when subject to the precessional loads incident to operation in three types of aircraft. The test demonstrated that the commercial "J"-type bearings used for this test were good for the equivalent of 750 to 1,000 hours of operation when used in aircraft similar to the XB-70. The test also demonstrated that when selectively located (i.e., flywheel axis of rotation parallel to fuselage centerline so that roll maneuvers do not generate precession loads) the bearings were good for 750 to 1,000 hours in more maneuverable aircraft. The latter test completed only 17 percent of the 10,000-cycle goal because the components available for the test were misconfigured for the application.

As a result of these tests numerous conclusions have been drawn. A number of these conclusions, dealing with a specific demonstration, have already been discussed at the end of the section covering that particular demonstration. Some of the more important of these will be repeated here. However, in general, the conclusions discussed in this section will cover those drawn from the continuous duty cycle tests and those drawn from the overall program. These conclusions are as follows:

1. Energy storage substations can be designed and fabricated to supply performance and weight saving improvements in excess of those which were established by analysis in the Phase I program (Report No. AFAPL-TR-66-29).

2. The high speed energy storage flywheel is compatible with the normal space, weight, reliability, service life, and maintainability requirements, as well as most of the environmental conditions which will be encountered in present-day and near-future aircraft.

3. The energy storage substation, which is inherently a source of mechanical shaft power, can be successfully coupled to a mechanical actuator. This approach is potentially the best method for utilizing the substation's full capability for both "intermittent" and "continuous"-type applications. However, before this full

capability can be realized, additional work must be done on mechanical couplings and particularly on the mechanical servo needed for continuous duty-cycle-type aplications.

4. Charging and discharging a flywheel energy storage device has been demonstrated to be practical using hydraulic means.

5. The speed range of the flywheel and a present-day motor-pump can be coupled through a single mesh of spur gearing. The necessary technology to build such gearing with long life capability at high speed is presently available without further need for development.

6. The energy storage substation concept will enhance survivability. This conclusion is derived from several basic characteristics of the system as follows:

   a. The energy storage substation reduces the size of the power generation and distribution system, thus reducing its target area with respect to enemy groundfire.

   b. The energy storage substation is adaptable to use in armored power packs close to the point of power usage.

   c. The energy storage substation is amenable to a variety of power inputs (hydraulic, pneumatic, electrical, or mechanical).

   d. If the power from the primary power source is interrupted for any reason (such as gunfire damage), the stored energy in the substation will give the air vehicle greatly enhanced "get home" capability.

   e. The flywheel containing energy storage substation is very amenable to mechanical power extraction. In its turn, mechanical power extraction when developed will make possible the construction of a compact, relatively projectile-resistant package.

The conclusions which can be drawn with respect to the various components making up the energy storage substation are as follows:

1. Motor Pump (New York Airbrake - Part No. 64 WK020152) - After it was found that this unit must operate at 150 to 200 psi case pressure, the performance of the motor-pump was consistently successful. Both the continuous duty test and the intermittent duty tests show that the principle of storing and extracting energy by means of a motor-pump is a very practical approach.

2. Gearbox (Vi-Star Gear Company - Model 6701) - This gearbox was operated at 8,000 rpm input and 52,400 rpm output and proved to be extremely rugged and reliable. Even though the gearbox was occasionally inadvertently abused, there was no failure chargeable to the gearbox. One of the gearboxes in the program has logged in excess of 1,576 hours and the other two range between 100 and 200 hours.

3. Flywheel (North American Rockwell Corporation - Part No. EMR 2655-107) - Except for the precessional load problem previously discussed, the flywheel has proved to be an extremely reliable device. Several interesting failure mode characteristics were uncovered as follows:

   a. In the event of bearing failure, the flywheel, acting as a gyroscope, continues to rotate around its own axis using the remaining bearing as support. The only sign of failure is rapid bearing temperature rise. There is very little audible sound level change and the flywheel will coast to a stop smoothly on shutdown.

   b. The flywheel tip is traveling at 132,000 feet per minute (mach 2.3) in an evacuated housing (2 cm Hg abs). It was originally feared that if a leak occurred and a lubricating oil-air mixture entered the housing there was danger of a fire or explosion. Several leaks of this type have occurred and, so far, the only effect has been flywheel slowdown and increases in flywheel housing temperatures approaching 350° F. In one instance, mercury was drawn into the housing from a manometer. The result was a rapid flywheel slowdown and degradation in flywheel tip strength from 260,000 psi tensile to 210,000 psi. Based upon this tensile strength decrease, it is believed that the tip temperatures exceeded 600° F. However, nothing untoward occurred and this flywheel is the one which continued to complete the continuous duty cycle endurance test.

   c. Vibration at critical speeds were considered a major problem which might lead to early flywheel failure. However, the flywheels used for this program were balanced to less than 0.005 inch-ounce and no critical speeds were encountered for any of the three flywheels involved.

4. Evacuation Scavenge and Lubricating Systems - These functions were accomplished by a compact stack of W. H. Nichols "Gerotor" pumps driven by a common drive shaft. All three systems worked without incident throughout the test in spite of the fact that, in at least two instances, large quantities of metal chips resulting from a flywheel bearing failure passed through the scavenge pump. The most interesting conclusion reached is the fact that a small gear pump will consistently maintain a vacuum of 2 to 3 cm Hg abs when its outlet is wetted with oil.

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13 ABSTRACT

Essentially there are two principal uses for flywheels. One is to create precessional forces (such as gyroscopes), the other is to store up kinetic energy. This program demonstrated the use of optimized energy storage flywheels for the purpose of supplying peak energy demands in aircraft flight control and utility actuation function duty cycles.

Three simulators were used. One was a tilt table for investigation of precessional loads on bearings. Another was the XB-70 landing gear simulator for demonstration of both hydraulic and mechanical power extraction from flywheels in the performance of intermittent type of aircraft actuation duty cycles. A third simulator demonstrated 1,500 hours of aircraft flight control operation, wherein a bank of three XB-70 elevons were supplied one third of their peak hydraulic flow from a motor-pump driven by a flywheel. This test demonstrated 300 flights of 5 hours each.

The program demonstrated the operating characteristics and the feasibility of using flywheel energy storage substations in the accomplishment of aircraft actuation functions. The program also verified the validity of the weight saving estimates made in Phase I.

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Energy Storage						
Gears						
Bearings						
Shaft Seals						
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There is no limitation on the length of the abstract. However, the suggested length is from 150 to 225 words.

14. KEY WORDS: Key words are technically meaningful terms or short phrases that characterize a report and may be used as index entries for cataloging the report. Key words must be selected so that no security classification is required. Identifiers, such as equipment model designation, trade name, military project code name, geographic location, may be used as key words but will be followed by an indication of technical context. The assignment of links, rules, and weights is optional.

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